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Internal combustion engines of modern marine vessels

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Textbook for maritime educational institutions

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Over the past 20-25 years, marine internal combustion engines have undergone significant changes. These changes are reflected in this textbook. At the same time, questions regarding the design and operation features of discontinued engines are excluded from consideration or minimized. Attention is paid to the issues of increasing the economic and environmental performance of marine engines by improving pressurization systems, fuel supply, the use of gas fuel, as well as by supplying water to the engine working space. The theoretical aspects of the processes occurring in modern engines of all types and sizes are considered.

This book will be useful to anyone who studies marine internal combustion engines or is involved in their design and operation.

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Євген Білоусов, Микола Булгаков

Двигуни внутрішнього згоряння сучасних морських суден

Підручник для морських навчальних закладів

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За останні 20-25 років суднові двигуни внутрішнього згоряння зазнали значних змін. Ці зміни відображені в цьому підручнику. При цьому питання щодо конструкції та особливостей роботи знятих з виробництва двигунів виключені з розгляду або зведені до мінімуму. Приділено увагу питанням підвищення економічних та екологічних показників суднових двигунів шляхом удосконалення систем наддуву, паливопостачання, використання газового палива, а також шляхом подачі води в робочий простір двигуна. Розглянуто теоретичні аспекти процесів, що відбуваються в сучасних двигунах усіх типів і розмірів.

Книга буде корисна всім, хто вивчає суднові двигуни внутрішнього згоряння або займається їх проектуванням і експлуатацією.

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CONTENTS

INTRODUCTION

Marine internal combustion engines have firmly taken leading positions in the global merchant fleet as the main and auxiliary energy sources that provide both the propulsion of the vessel and the generation of electrical and other types of energy used to support technological processes, the life of the crew and passengers. Due to the peculiarities of the organization of the operating process, in which fuel combustion occurs directly in the working cylinders, internal combustion engines today have the highest efficiency in converting thermal energy into mechanical work. In addition, the use of heavy fuels obtained as residual products of oil refining in marine diesel engines has significantly reduced the cost of generated energy. The introduction of new technologies related to the use of electronic control systems, as well as the transition to gas and gas-diesel cycles, has made it possible to successfully address issues of increasing the environmental performance of ship engines. Internal combustion engines are relatively simple and technologically advanced to operate, and have a long service life, often commensurate with the service life of the vessel on which they are installed.

All of the above has led to the fact that today 98% of the world's merchant fleet ships are equipped with internal combustion engines. Marine engine building is a leading sector of the world economy, which is dynamically developing and strives to respond to market demands as quickly as possible.

Currently, marine internal combustion engines are produced by several dozen manufacturers who have their own approaches to solving certain problems related to increasing fuel efficiency, environmental friendliness, reliability and reducing operating costs. Generalization of the accumulated experience in solving these problems can be useful to specialists engaged in both the design and operation of engines, as well as undergraduate and graduate students studying in maritime educational institutions. It is for this audience that the book offered to your attention is intended. The information provided in the book is accompanied by drawings, drawings and other graphic material that give an idea of the design solutions used by one or another manufacturer.

It should be noted that the volume and structure of information provided by different manufacturers may vary greatly, so the data presented in the book may differ in the completeness of presentation. The information provided in the book can only be used for preliminary acquaintance with the main design solutions and theoretical foundations of the processes occurring in engines. To obtain more detailed information, please refer to specialized literature.

The information provided in the book maximally corresponds to the requirements for the training of ship mechanics, both at the operational and management levels, set out in the International Convention on Standards of Training, Certification and Watchkeeping for Seafarers, 1978.

Based on this, the main purpose of the book is its use in the educational process of maritime educational institutions that train maritime specialists for the merchant fleet. At the same time, the book will be useful to existing mechanics, as well as to everyone involved in the design and operation of marine piston internal combustion engines.

The authors express special gratitude to the reviewers for a number of valuable comments made during the preparation of the manuscript. The authors also express gratitude to the teams of the Department of Operation of Ship Power Plants of the Kherson State Maritime Academy and the Department of Ship Power Plants and Technical Operation of the Odessa National Maritime University for their assistance in collecting and preparing materials.

SECTION 1

History of piston internal combustion engines

The history of reciprocating internal combustion engines (ICEs) dates back to 1678. In this year, the first piston engine was proposed by the outstanding Dutch mathematician, mechanical engineer and physicist Christian Huygens (1629-1695). It was assumed that black powder would be used as fuel for such an engine. However, Huygens' proposed engine was never built.

Proposed in 1860 by the Belgian engineer Etienne Lenoir* (1822-1900), the engine worked without compression and its efficiency was slightly different from steam engines, so it did not compete significantly with them. However, given its relative simplicity due to the absence of a steam boiler, this engine became quite widespread.

The idea of organising the operating process with pre-compression of the working medium, expressed as early as 1828 by Sadi Carnot*, was later implemented by Eugene Langer* (1833-1895) and Nicolos Otto* (1832-1891) for the four-stroke engine they designed. This engine consumed four times less fuel than Lenoir's engine and was the first efficient reciprocating internal combustion engine.

The first internal combustion engines ran on generator gas, which created certain inconveniences for their use, especially in transport. As a result of numerous searches for universal fuels for internal combustion engines, the choice fell on petroleum products. This type of fuel was further associated with a significant improvement of Otto engines and their widespread use in transport, but the efficiency of such engines remained quite low for a long time.

Attempts to create a piston internal combustion engine with high fuel efficiency were made by many engineers of that time, but the greatest success was achieved by the German engineer Rudolf Diesel* (1858-1913) (Fig. 1.1). On 28 February 1892, he was granted patent No. 67207 for «Operating process and Method of Execution of a Single-Cylinder and Multi-cylinder Engine». About a year after the patent was granted, Diesel's brochure entitled «Theory and Construction of a Rational Heat Engine to Replace the Steam Engine and Other Presently Existing Engines» was printed by the technical literature publisher Julius Springer in Berlin. In this brochure, Diesel, analysing the experience of creating piston engines, as well as the basic theoretical provisions of thermodynamics, including those developed by the founder of technical thermodynamics Sadi Carnot* in his work «On the motive power of fire and on machines capable of transforming this power», proposed the concept of creating a rational heat engine.

It is interesting to note that Diesel originally intended to use fine coal dust as fuel for his engine, which was to be fed into the cylinder at the end of the compression process by a jet of compressed air. Diesel was convinced that the energy of the air charge, as a result of high compression, could exceed the activation threshold of the fuel combustion process and, as a consequence, self-ignition would occur in the engine cylinder. For this purpose, it was proposed to significantly increase the compression ratio of the air charge compared to the Otto engine.

Diesel's work aroused great interest among specialists, finding both supporters and opponents. In fact, a fundamentally new type of engine with gradual combustion of fuel atomised in an air charge was proposed. Many assessed the provisions outlined in the brochure as extremely interesting in theoretical terms, but hardly feasible in practice. Nevertheless, there was no doubt in anyone's mind that, if the theoretical premises described by Diesel could be realised in practice, the engine thus obtained would have unquestionable advantages over all heat engines known at the time.

* brief information about this figure of science and technology is given at the end of the book

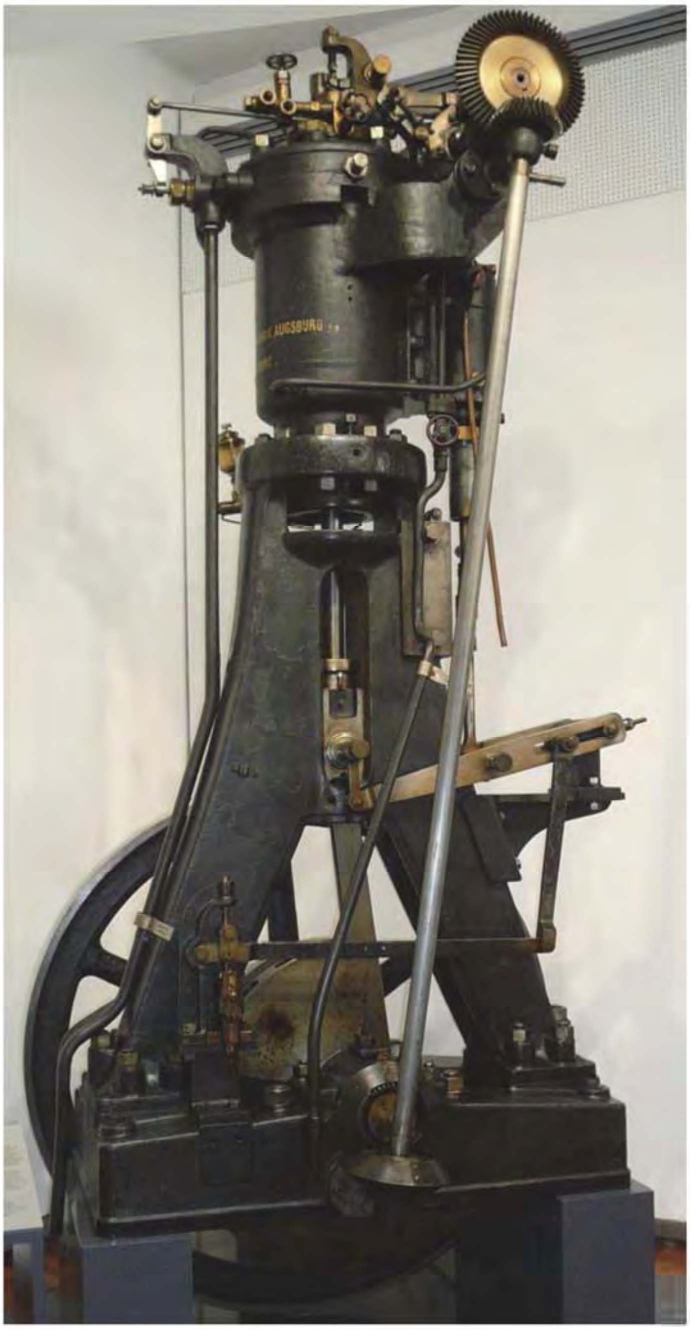

Figure 1.2 – One of the first workable R. Diesel engines [1]

Therefore, even in this form, the Diesel engine began to enjoy commercial demand, and the main efforts of the inventor, as well as most other engineers, were directed at improving the operating process based on the use of liquid fuels.

Among those who recognised the prospects of the new invention for use as a marine engine was Joan Jacob Sulzer* (1855-1922), a descendant of the founders of the Sulzer Brothers* company, which at that time was manufacturing marine steam engines in Winterthur, Switzerland. In 1879, R. Diesel, while still a student at the Technical High School in Munich, had an internship at this company, and with the son of the founder of the company he was connected not only by business relations, but also by personal friendship. In 1898 the Sulzer brothers' firm acquired the engine patent and during the next few years completely reoriented its production to the production of new products. One of the first marine diesel engines produced at this company is shown in Fig. 1.3. In 1905, the Sulzer brothers' factory was the first to design two-stroke engines, which are now predominantly used in the merchant marine.

The owners of the Danish engineering concern Burmeister & Wain* had a similar foresight. They acquired a patent in 1898 and started production of stationary and marine Diesel engines. The first engine of this concern has survived to the present day (Fig. 1.4).

SECTION 2.

General information on marine internal engines of combustion engines and their operation

2.1 General structure and principle of operation of marine piston internal combustion engines

Diesel internal combustion engines (hereinafter referred to as diesels) are predominantly used in the fleet. They are installed on 95...98% of sea and river vessels. Other types of engines are rarely used on merchant ships, and their consideration is beyond the scope of this course. Therefore, hereinafter the term internal combustion engine (ICE) or marine internal combustion engine (MICE) will be used to refer to a piston engine of the diesel type.

An internal combustion engine is a reciprocating heat machine in which the chemical energy of the fuel is first converted into heat, and then the heat energy is converted into mechanical work.

All processes related to the conversion of fuel energy into mechanical work in this class of heat engines take place in the space of the working cylinder. Heat is supplied to the working medium in the process of fuel combustion in the combustion chamber. At the first stage the working medium is air (filling and compression processes), and at the second stage (after ignition and combustion of fuel) – gaseous products of fuel combustion, which, expanding, do useful work.

This method of energy conversion has significant advantages over other types of heat machines, as the hot heat source is located inside the engine itself. As a result, there are no losses associated with heat exchange, as is the case, for example, in the cycles of steam power plants. As a result, the engine is compact, as there are no bulky and inefficient heat exchangers.

The rapidity of the operating process makes it possible to trigger higher temperature differences in the internal combustion engine without damaging the structural elements of the engine itself. This in turn leads to an increase in the thermal and effective efficiency of the cycle. Today marine diesel engines are the most efficient heat engine available to mankind. The best samples of such engines have an efficiency of 48...54%, and with heat recovery systems this figure is even higher.

All modern internal combustion engines are designed to force high-pressure air into the engine cylinders. This method of air supply is called supercharging. The use of supercharging provides higher mass filling of the cylinders, which makes it possible to increase the cycle fuel supply (compared to un-supercharged internal combustion engines) and, consequently, to obtain greater effective power at the engine crankshaft output flange.

The most common is gas turbine supercharging, where the energy released by the expansion of exhaust gases on the blades of the gas turbine is used to drive a centrifugal compressor, which precompresses the air to the required parameters. Such supercharging units are called gas turbochargers.

Studies of the operating process have shown that this method of supercharging allows, through the use of exhaust gas energy, to increase the efficiency of the diesel engine, its economy and the power removed from the crankshaft output flange.

In the case of two-stroke engines, the introduction of a gas turbine supercharger allowed to simplify the purge scheme considerably, eliminating the need for bulky purge pumps. The combination of a piston engine and a gas turbocharger in one power unit affects all thermal and gas-dynamic processes in the engine cylinders.

An internal combustion engine is a piston-type thermal machine in which the chemical energy of the fuel is first converted into heat, and then the heat energy is converted into mechanical work.

Engines combining a piston machine, gas turbine and compressor in one unit are called combined engines. The general structure of a modern marine two-stroke engine with gas turbine supercharging is shown in Fig. 2.1.

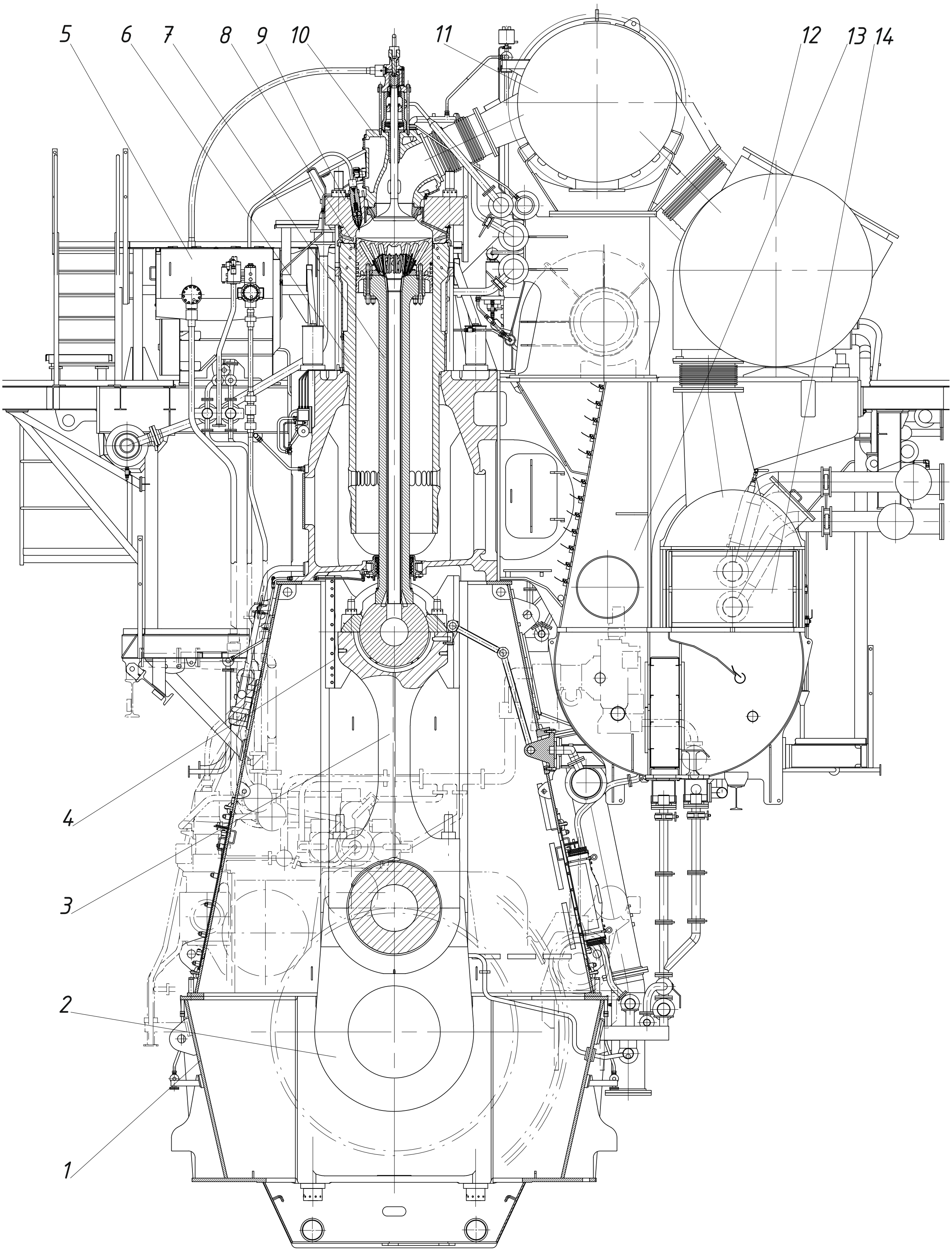

Figure 2.1 – General structure of the RT-flex 82C two-stroke crosshead diesel engine: 1 – frame; 2 – crankshaft; 3 – connecting rod; 4 – crosshead; 5 – fuel supply and gas distribution control unit; 6 – cylinder sleeve; 7 – piston rod 8 – piston; 9 – cylinder cover; 10 – exhaust valve actuator; 11 – exhaust receiver; 12 – turbocharger; 13 – intake receiver; 14 – charge air cooler (adapted from [1])

Bottom dead centre (BDC) – is the position of the piston at which the distance from the piston to the crankshaft axis is smallest.

The cylinder diameter D – is the main geometric dimension of the engine and the choice of other engine dimensions depends on it to a large extent.

Piston stroke S is the distance travelled by the piston from one dead centre to another. In the central mechanism, each piston stroke corresponds to a 180° rotation of the crankshaft.

The *crank radius r* – is the distance between the frame and crank journals of the crankshaft. For a centre crank mechanism, the piston stroke $S = 2r$ (for each specific engine, S and r are constant values).

The volume of the compression chamber or combustion chamber V_c, is the volume of the cylinder above the piston at TDC.

The working volume of a cylinder V_s, is the volume displaced or released by the piston in one stroke from BDC to TDC and vice versa:

$$V_s = (\pi D^2 S)/4.$$

The total cylinder volume V_a, is the volume above the piston at its position at BDC.

$$V_a = V_s + V_c$$

The length of the connecting rod l_{cr}, largely determines the height of the engine, as well as the nature and magnitude of the acting forces and moments in the crank mechanism.

The theoretical or *geometric compression* ratio ε is the ratio of the total cylinder volume to the volume of the compression chamber:

$$\varepsilon = \frac{V_a}{V_c} = \frac{V_s + V_c}{V_c} = 1 + \frac{V_s}{V_c}.$$

The compression ratio is a dimensionless value that shows how many times the volume of the cylinder above the piston decreases, i.e. the charge in the cylinder is compressed, when the piston is moved from the BDC to the TDC.

The theoretical compression ratio is used for approximate calculations of operating processes of four-stroke engines, in which it can be conventionally considered that the compression process begins at BDC (except for four-stroke engines operating on the Miller cycle).

In all types of engines, with increasing compression ratio, heat utilisation improves and, consequently, the indicator efficiency of the engine increases (Fig. 2.5), which has a positive effect on their performance characteristics. However, at the same time at medium and maximum loads, the toxicity of exhaust gases increases (the amount of hydrocarbons due to the increase in the volume of the wall layer during combustion and the amount of nitrogen oxides due to the increase in combustion temperature). The load on the crank-rod mechanism also increases, and in order to ensure reliable operation of the engine it is necessary to increase the size and weight of its main parts accordingly. As a consequence, mechanical losses increase, which gradually offset the advantages associated with the increase in thermal efficiency (Fig. 2.5).

In four-stroke diesel engines, as the compression ratio increases, the temperature and pressure at the time of fuel injection start increase, resulting in a lower pressure build-up rate ($dp/d\varphi$) and softer engine operation. Increasing ε also improves the starting properties of the engine. Indicator efficiency in the range of compression ratios (ε = 14.5...19) used in diesel engines changes insignificantly. Temperature increase, at large values of ε, leads to an increase in the content of nitrogen oxides in the combustion products.

In practice, the lower limit ε, is determined by the need to achieve in the working cylinder at the end of compression temperature providing guaranteed autoignition of the fuel supplied, the upper limit – a rational ratio of thermal efficiency and mechanical losses (shaded area in Fig. 2.5).

In two-stroke engines, due to the specifics of the operating process, part of the working stroke of the piston is lost to organise gas exchange between the working cylinder and the environment. To remove combustion products and supply fresh air, the cylinder has special purge ports or ports

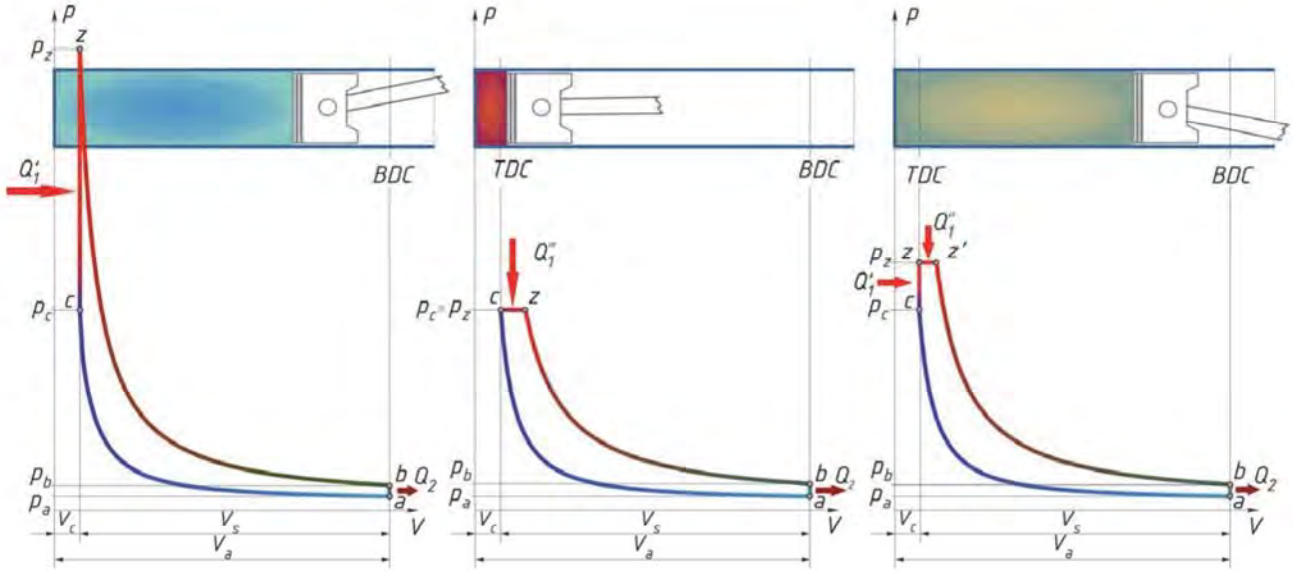

Figure 2.7 – Ideal piston engine cycles. *a* – Otto cycle; *b* – Diesel cycle; *c* – Trinkler-Sabathé cycle

The gas pressure force P_p is equal to the product of the average pressure in the cylinder p_p at the n-th section of the cycle by the piston area F_p

$$P_p = p_p F_p.$$

Substituting into the expression for the work at the n-th section of the cycle, we obtain the elementary work:

$$L_c^n = p_p F_p S_p.$$

Considering that $F_p S_p = V_s^n$, i.e. equal to the elementary volume at the n-th section of the cycle, finally the formula of work for the elementary section will take the form:

$$L_c^n = p_p V_s^n,$$

The total work of the cycle is equal to the sum of the work in the elementary sections:

$$L_c = \sum_1^n L_c^n$$

Thus, the area of the figure bounded by the processes constituting the cycle in pV coordinates will be equivalent to the work of the cycle.

The Trinkler-Sabathé* cycle impoverishes the features of the Otto* and Diesel cycles, so the main conclusions for this cycle are to some extent true for the other two cycles as well. Further we will consider these regularities in more detail.

In a cycle with mixed heat input, *as* – is the adiabatic compression of an ideal gas; *czz'* – is the heat input at V=const and p = const (in a real engine this corresponds to the process of fuel combustion), *z'b* – is the adiabatic expansion of the gas, *ba* – is the heat removal to a cold source at V= const.

The thermal efficiency of a mixed heat input cycle, all other things being equal, occupies an intermediate position between the first and second cycles. In general, it can be expressed as:

$$\eta_t = \frac{(Q_1' + Q_2'') - Q_2}{Q_1' + Q''} = 1 - \frac{Q_2}{Q_1' + Q''}$$

Expressing the heat input $Q_1 = Q'_1 + Q''_1$ and the heat output Q_2 through the temperatures at characteristic points of the cycle and the heat capacities and making the appropriate transformations, we obtain the formula of Seileger M.P.*:

$$\eta_t = 1 - \frac{1}{\varepsilon^{k-1}} \frac{\lambda\rho^k - 1}{\lambda - 1 + k\lambda(\rho - 1)} \tag{2.1}$$

2.4 Real cycles of marine diesel engines

The four-stroke engines used in the navy have predominantly valve timing with the valves mounted on top of the cylinder head. The valves are held closed by springs and by the pressure of the gases in the cylinder. The valves are opened at the right moments by a timing mechanism driven by the engine crankshaft. Each valve is opened once per two revolutions of the crankshaft.

For both ideal and real cycles, the engine operating processe is traditionally represented in pV-coordinates. For a real cycle, auxiliary processes associated with the need for gas exchange between the working cylinder and the environment are additionally plotted on the diagram. Schematic representation of the four-stroke diesel engine operation order combined with indicator diagrams and diagrams of gas distribution phases is presented in Fig. 2.8.

Timing diagrams – represent the duration of the processes occurring in a real engine, which are represented in the polar coordinate system as a function of the crankshaft rotation angle. The pole of the diagram is conventionally aligned with the crank rotation axis, and the processes are represented as circular arcs.

2.4.1 Operating procedures for a four-stroke diesel engine

The first stroke* is *intake (Fig. 2.9, *a*). When the crankshaft rotates (in the direction of the arrow), the piston moves from TDC to BDC, and the distributor mechanism opens the intake valve 1, and communicates the cylinder supra-piston space with the intake pipe.

In a real engine, the intake process consists of three stages.

The first stage (I) is the dynamic purging stage. The inlet valve opening starts 50...80° before the piston reaches TDC (point r on the indicator diagram), this interval is called the *inlet valve opening advance angle* (φ_{Iiv}). This is done to ensure that the cylinder is filled to its maximum capacity with air, so that there is sufficient air flow through the cylinder by the time the piston starts to move to TDC. The combustion chamber volume is dynamically purged by the energy of the exhaust gases and the air pressure in the intake receiver.

The second stage (II) is the mechanical suction stage. The piston moves from TDC to BDC, increasing the cylinder displacement. Through the open inlet valve, the turbocharger (TC) supplies air into the super-piston space under pressure p_k =0.13...0.45 MPa.

The third stage (III) is the stage of dynamic supercharging. After the piston arrives at BDC, the intake valve does not close immediately, but remains open for another 30...50°. This angular interval is called the *intake valve closing lag angle* (φ_{IIIiv}). During this interval of the cycle, the cylinder is dynamically charged with air due to the inertia of the gas flow in the intake pipe, the velocity of which can reach 70 m/s by the end of intake.

On the indicator diagram the intake process corresponds to the line *ra.* At the end of the filling process, the air in the cylinder has the following parameters:

$$p_a = 0.130...0.45 \text{ MPa}; \; t_a = 40...130°\text{C}.$$

The second stroke is compression. The piston moves from BDC to TDC, compressing the air charge (Fig. 2.9, *b*).

In a real engine, compression starts from the moment of intake valve closure (point a on the indicator diagram) and continues until the piston reaches TDC. By the end of the compression process, the charge temperature must ensure stable autoignition of the fuel, for this purpose it must exceed the autoignition temperature of the fuel by 160...200°C.

An ideal compression process (line 1 in Fig. 2.10) should be adiabatic and described by the equation pV^{k_1} = const, where k_1 – is the compression adiabatic exponent. In the real compression process, there are always leakages of charge through leakages in the working space and heat exchange with the cylinder walls.

Indicator efficiency takes into account not only heat removal to the cold source, but also heat losses in the working cylinder. It characterises the degree of perfection of conversion of heat energy into mechanical work inside the engine cylinder.

ter of the process with the index n_2 (purple dashed line – Fig. 2.11), which varies depending on the intensity of the prevailing factors throughout the process.

In the first part of the process (up to the point p_z), the dominant factors are heat input, molecular changes and intensive heat exchange with the cylinder walls. In this section, the value of the expansion polytropy index n_2 is lower than the expansion adiabatic index k_2, in some cases even less than 1. It should be noted that molecular change in the composition of the working medium leads to a decrease in the adiabatic index, until the process of fuel afterburning is completely completed on the expansion line at point 2 (Fig. 2.11). As the piston moves to the BDC, the process of fuel afterburning becomes less intensive, and the heat exchange with the walls increases due to the increase in the cooling surface. At this point, the polytropic index of expansion increases continuously.

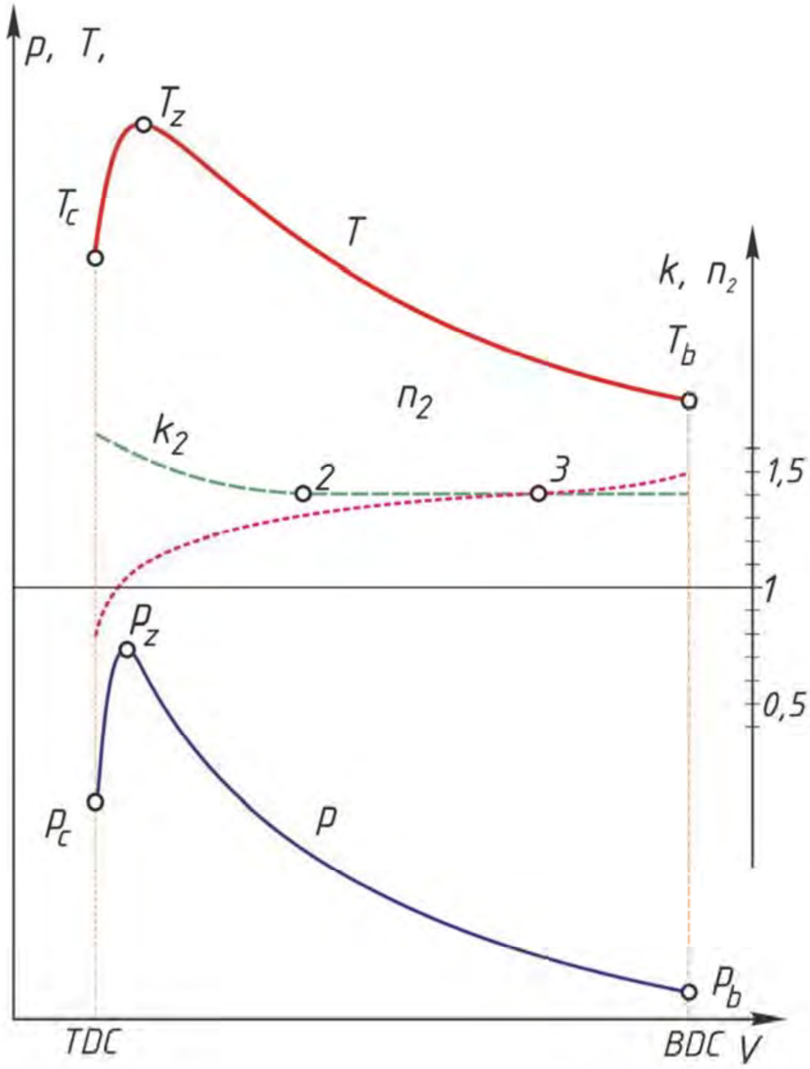

Figure 2.11 – Change in the process of expansion of indicators T, p, n_2 and k_2

At a certain position of the piston, the heat input due to afterburning of fuel and recovery of dissociation products becomes equal to the heat transfer to the cylinder walls. The current value of the polytropy index reaches the value of the adiabatic index, $n_2 = k_2$ (point 3 – instantaneous thermal equilibrium of the state of gases and cylinder walls). At further expansion the heat dissipation into the cylinder walls becomes the dominant factor, and the index n_2 becomes higher than the adiabatic index k_2. As the piston approaches the BDC n_2 increases to 1.4...1.5. Thus, in a real engine, the expansion process after completion of visible combustion is carried out by polytrope with a variable index from 1.1 to 1.5.

Pressure and temperature of the end of expansion. *The end expansion* parameters (pressure p_b and temperature T_b of the gases) depend on the maximum combustion parameters $p_{z'}$, $T_{z'}$, the degree of subsequent expansionδ and the expansion polytropy n_2:

- the exhaust and intake process is controlled by the piston itself as it moves around the BDC.

In this type of engine, the piston, as it moves from TDC to BDC, first opens the exhaust ports, through which gases enter the exhaust receiver and then the turbine. As the piston moves further, the purge ports open and air pre-compressed in the turbocharger enters the cylinder. The air supply is organised in such a way that the exhaust gases are cleared from the cylinder as much as possible.

The timing of these engines is symmetrical with respect to the TDC and cannot be changed during engine operation.

The advantages of this purging scheme are:

- simplicity of engine design, which is expressed in the absence of a complex system of valves and their drive, as well as a simpler design of the working cylinder head;
- the possibility of centralised injectors positioning, which promotes quality mixing.

The disadvantages of engines of this design include:

- significant loss of operating stroke due to the high height of the exhaust ports;
- impossibility of quality cleaning of the working space from exhaust gases. There are always areas in the cylinder that are practically free of fresh charge;
- the need to have sufficient piston height to avoid simultaneous opening of the purge and exhaust ports when the piston is close to TDC;
- high charge losses due to the fact that from the moment of closing the purge ports to the moment of closing the exhaust ports, the charge is displaced by the piston into the exhaust receiver.

For the above reasons, which do not allow further forcing of the operating process, the slotted purge schemes are now abandoned. The last manufacturer of slotted purge engines was the Swiss company Sulzer, for which this engine layout was traditional. However, since 1981, the company has started production of new RTA series engines, which use a straight-through valve-slotted purge arrangement.

2.5 Main engine performance indicators

2.5.1 Indicator indicators

As it was shown earlier, the work done by gases in the working cylinder is equivalent to the area of the indicator diagram. In practice, it is rather difficult to consider the area of the curvilinear diagram, so for simplicity of processing it is represented in the form of a rectangle, one side of which is equal to the working volume V_s, and the other side corresponds to some average pressure, under the action of which the piston during one working stroke does work equivalent to the indicator work of a real closed cycle. Such conditional pressure has received the name of *average indicator pressure* – p_i, and the work done per cycle, *indicator work* – L_i.

To find p_i, the area of the indicator diagram is represented as the sum of rectangles equal in area to the elementary fragments of the diagram, the curvature of which is neglected. The total indicator work of the cycle L_i is found as the sum of elementary works, kJ (Fig. 2.14):

$$L_i = \sum_{n=1}^{n=20} L_n ,$$

and the average indicator pressure will be equal to, kPa (Fig. 2.15),

$$p_i = L_i / V_s. \tag{2.6}$$

If we refer the work of the cycle to the working volume of the cylinder in m^3, we obtain the specific work of the actual cycle in the form:

$$p_i = \frac{L_i}{10^3 V_s}, \text{ kJ/m}^3.$$

Thus, p_i defines the cycle work attributed to one m^3 of cylinder displacement and is independent of its geometrical dimensions.

in the engine mechanisms, to drive auxiliary mechanisms that ensure its operation, to perform pumping strokes of the piston and others. Power losses associated with the above factors are called the power of *mechanical losses* – N_m. If this power is divided by the working volume of the cylinder, we get a value called the *mechanical loss pressure* $p_m = N_m / V_s$.

Friction losses, as a rule, make up the major part of mechanical losses (55...65%). They are caused by friction in all mating friction pairs, the main ones being friction of piston rings of the piston and cylinder sleeve due to high specific pressure of the rings on the sleeve, as well as friction in crankshaft bearings.

The effective power is the power that can be removed from the engine crankshaft flange – N_e. It is equal to:

$$N_e = N_i - N_m.$$

Mechanical efficiency. The ratio of effective power to indicator power is the *mechanical* efficiency of the engine:

$$\eta_m = N_e / N_i = (N_i - N_m)/N_i.$$

Relative mechanical losses depend on the design, speed and boost level of the engine. Mechanical efficiency characterises the degree of perfection of the engine mechanical design. Its values at rated power mode (usually η_m values are given for 100% engine load mode) are for:

- low-speed two-stroke diesel engines $\eta_m = 0.87...0.94$;
- medium-speed diesel engines $\eta_m = 0.84...0.92$;
- of high-speed diesel engines $\eta_m = 0.75...0.85$.

Average effective pressure is a conditional constant pressure p_e, which during one stroke of the piston would do work equal to the useful work on the engine crankshaft during one operating cycle. It characterises the average specific work of the cylinder per cycle and is one of the most important indicators characterising the engine load, the completeness and dynamics of combustion, the quality of the supercharger and the perfection of the design as a whole. This indicator is used to compare the load boost levels of different engines or different loading modes for the same engine.

The mean effective pressure can be expressed as (Fig. 2.11):

$$p_e = p_i - p_m.$$

If the values of mechanical efficiency and average indicator pressure are known, the value of p_e can be found as:

$$p_e = p_i \eta_m$$

Knowing the average effective pressure p_e it is possible to find the effective power of the engine, which is diverted to external consumers, kW:

$$N_e = \frac{p_e V_s n i}{60z} = \frac{p_e S F n i}{60z}.$$

The effective power N_e takes into account the mechanical losses in the engine in addition to the thermal losses, just like p_e:

$$N_e = N_i - N_m = \left(1 - \frac{N_m}{N_i}\right) N_i = N_i \eta_m, \text{ kW}.$$

The values N_e and p_e are the basic energy values of the engine as stated on the engine form.

The effective efficiency η_e characterises the efficiency of the final conversion of thermal energy Q_1 into mechanical work L_e on the engine shaft.

$$\eta_e = L_e / Q_1.$$

If we take, by analogy with the indicator work the effective efficiency:

TS-coordinates. The figures show that at the same pressures at the key points of the cycle lower temperatures operate at the same pressures, which indicates a decrease in the thermal stress of the operating process.

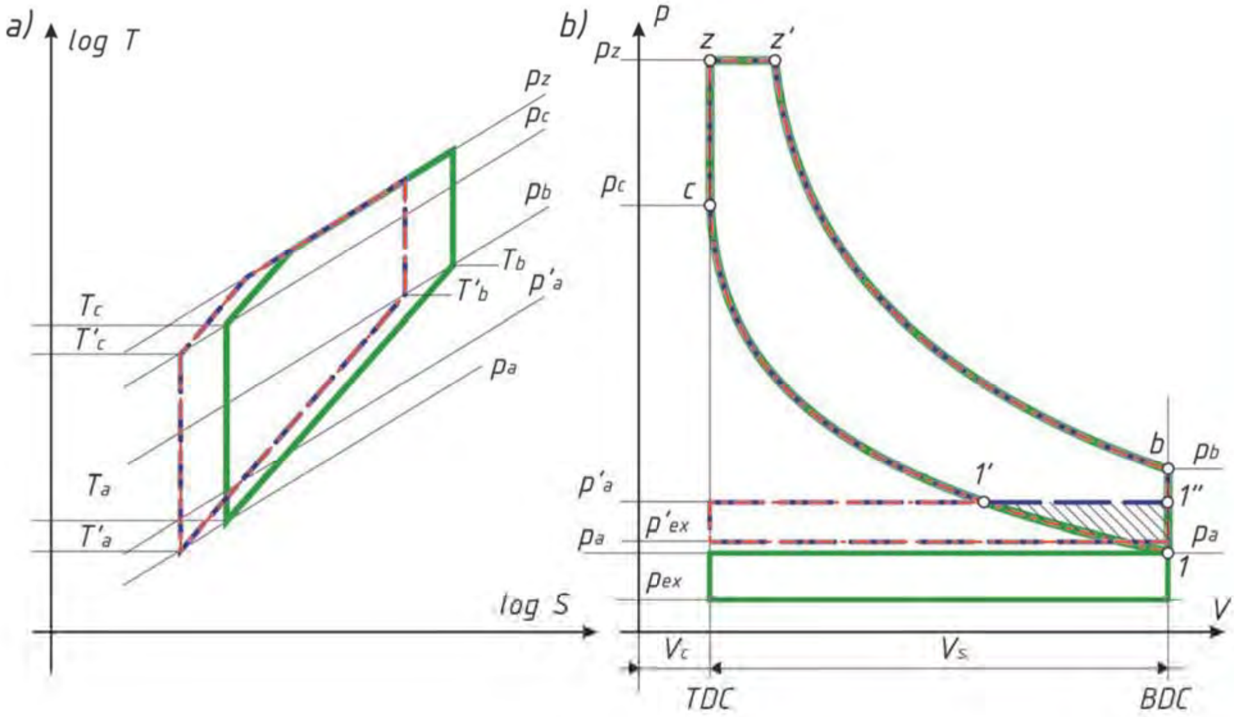

Figure 2.19 – Combined diagrams of operating processes of the basic engine and the Miller cycle with shortened compression and intake in *TS*-coordinates (*a*) and in *pV*-coordinates (*b*): ——— – basic cycle; – – – – – – with shortened compression; – – – – – – with shortened intake (adapted from [3, 4])

If we analyse theoretical cycles of engines operating under the Miller cycle, we can conclude that, all other things being equal, their efficiency will be lower than in the basic cycle due to the loss of part of positive work in the gas exchange section between the compression line and BDC (shaded area in Fig. 2.19 *b*). The loss of part of useful work in the gas exchange section, called «Miller losses», can be as high as 0.5 per cent, all other things being equal. In real cycles «Miller losses» are compensated, due to higher supercharging pressure and lower heat losses associated with reduction of the overall heat intensity of the operating process.

While maintaining boost pressure, switching to the Miller cycle results in lower pressures at key points in the cycle, resulting in lower mechanical stress, which in turn results in higher mechanical efficiency of the engine.

In order to quantify the influence of the selected gas distribution phases on the character of the operating process, a dimensionless indicator called the Miller coefficient is used:

$$m = S_M/S$$

where S_M is the part of the piston stroke in the Miller cycle spent on compression; S is the full stroke of the piston.

On this basis, in Miller cycle engines it is necessary to separate the concepts of compression ratio and expansion ratio. The expansion ratio is determined from the relationship $\varepsilon = (V_c + V_s)/V_c$ (Fig. 2.19 *b*), while the compression ratio depends on the Miller ratio and is determined from the relationship $\varepsilon_M = m(\varepsilon - 1) + 1$.

In the most general form, the expression can be used to assess the influence of the Miller coefficient on the efficiency of the operating process, all other things being equal:

$$\Delta\eta = \Delta\eta_{cce} + \Delta\eta_{egp},$$

where $\Delta\eta_{cce}$ – is the closed loop efficiency; $\Delta\eta_{egp}$ – efficiency of the gas exchange process.

Figure 2.20 shows the effect of operating processe efficiency components on the overall cycle

changes in the design of such engines, which will be discussed further in the relevant chapters, their operating process has undergone radical changes. This is primarily due to the fact that heavy fuels are characterised by a sufficiently high activation energy, which means the amount of heat that must be supplied to the fuel before it reacts with oxygen in the air.

Taking into account that the time for fuel combustion in medium- and high-speed engines is an order of magnitude shorter than in low-speed engines (in LSE at 100 min^{-1} 80...100 ms are allocated for the combustion process, in LSE at 1000 min^{-1} 5...8 ms), for qualitative combustion of fuel the activation energy should be supplied as soon as possible. The transition from heat supply to fuel to heat release as a result of reactions with air oxygen is called the activation threshold of the combustion process, and all processes preceding combustion are called pre-flame. To increase the rate of heat supply to the fuel, in order to reduce the time of pre-flame processes, it is possible to increase the temperature of the air charge and reduce the size of the fuel aerosol droplets. In practice, the former is achieved by increasing the temperature at the end of compression, and the latter by increasing the fuel atomisation pressure. To increase the temperature at the end of T_c compression to 900...1200 K, in this kind of engines, the compression ratio and boost pressure are increased. As a result, the pressure at the end of compression p_c, can reach 12...17.5 MPa, which is on the boundary close to the maximum allowable pressures based on the strength limits of the materials used. In this connection combustion in the engine working space should take place without significant pressure increase.

This is achieved by shifting the combustion process from the region close to the TDC to the expansion line. Theoretically, such a cycle is closer to the ideal Diesel cycle than to the Trinkler-Sabathé cycle. The shift of the combustion process to the expansion line is achieved by reducing the injection advance angle, as well as by using special fuel supply laws, including staged multiple injection.

Fig. 2.24 shows the pressure change as a function of the rotation angle in a conventional diesel engine operating process and in the Miller cycle with the combustion process shifted to the expansion line. The degree of pressure rise λ in such engines usually lies within 1...1.3. To achieve the specified parameters of the working medium by the end of compression, the boost pressure is increased to 0.3...0.4 MPa and the compression ratio to 13...16.

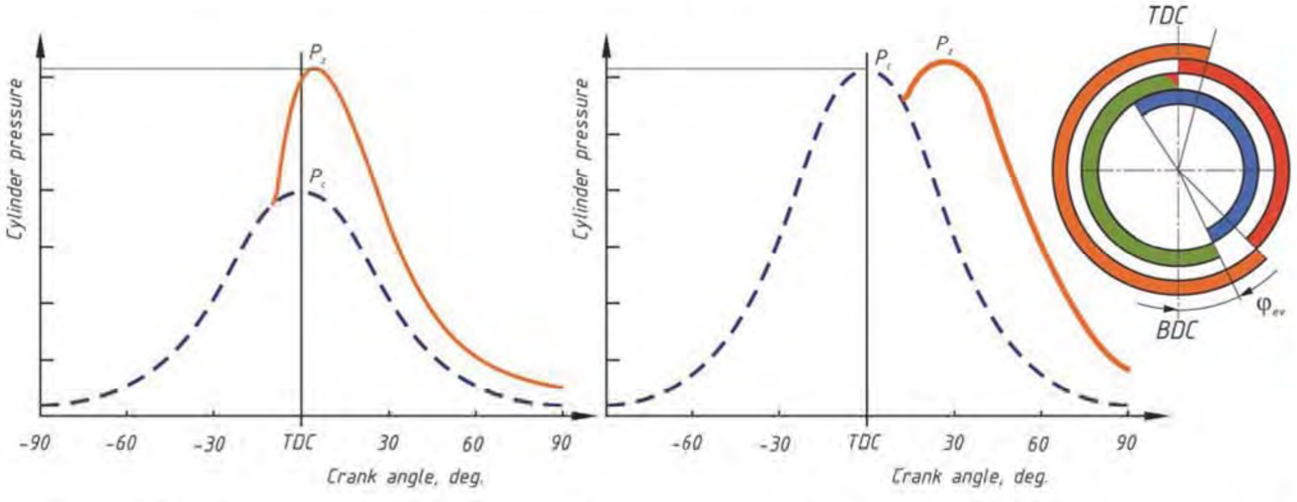

Figure 2.24 – Pressure variation in the operating processe as a function of crank angle. *a* – conventional operating processe; *b* – Miller process. – – – – compression line; —— combustion line; *c* – gas distribution phase diagram for Miller process

Decrease in thermal efficiency of the piston part of the engine associated with an increase in the degree of pre-expansion ρ is compensated by the use of turbochargers with a higher efficiency capable of effectively working with a higher temperature difference.

The activation energy of the combustion process is the amount of heat that must be supplied to the fuel to initiate reactions between the fuel molecules and the oxygen in the air.

nal surfaces of the engine.

As a result, all other things being equal, engines of this type have a shorter service life than crosshead engines. However, in newly developed diesel engines, a number of design solutions, which will be discussed below, allowed to bring the life of tronic engines to the same order of magnitude as low-speed crosshead engines.

The most characteristic layout of the trunk engine is an in-line arrangement of cylinders (Fig. 2.27 *a*). Single-row engines are simple in design and sufficiently technological in manufacture. These advantages, as well as a great experience of creation and operation of engines with vertically arranged cylinders cause their wide application in ship power engineering. Such arrangement allows to simplify enough the maintenance and repair of engines in the engine room. In particular, removal of heads, removal of pistons, etc. can be carried out using standard overhead hoisting and transporting mechanisms without additional accessories. The location of the various units of the engine systems on the sides of the crankcase makes it quite easy to monitor them during operation, service and repair. In-line engines are usually produced with 3 to 9 cylinders.

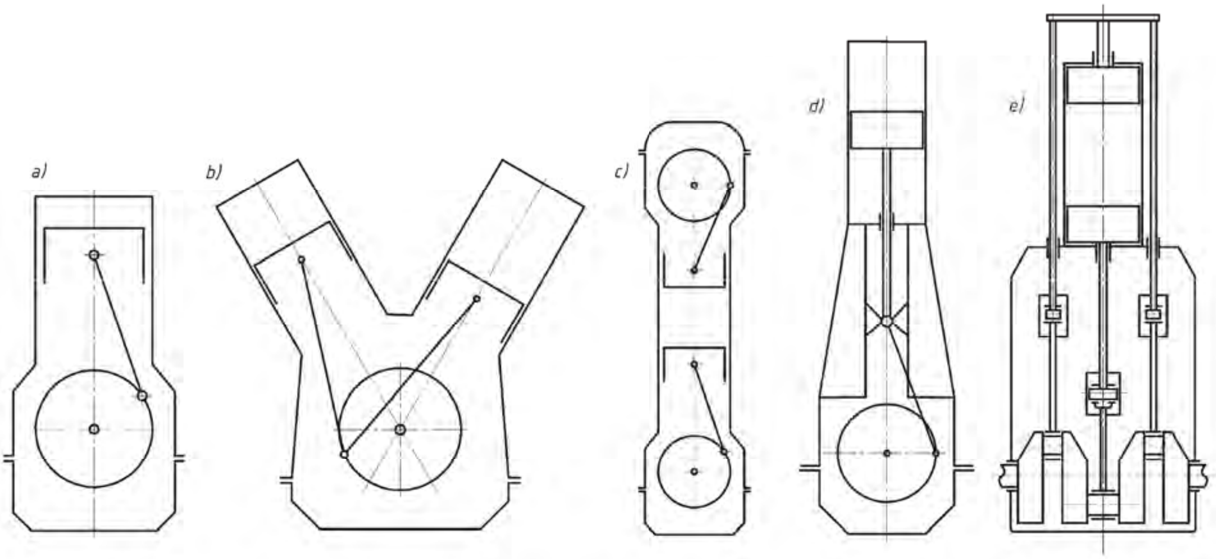

Figure 2.27 – Architectural forms of marine internal combustion engines: *a* – trunk engines in-line; *b* – trunk engines V-shaped; *c* – trunk engines with oppositely moving pistons; *d* – crosshead piston engines; *e* – crosshead engines with oppositely moving pistons

If it is necessary to increase the specific power, a two-row arrangement of working cylinders is used (Fig. 2.27 *b*), placing them relative to each other at an angle of 30...120°. This arrangement of cylinders is called V-shaped (after the shape of the Latin letter V). It allows, practically at the same power to halve the length of the engine and increase the rigidity of such critical structural elements as the frame and crankshaft. The angle between the cylinder axes, called the camber angle, is determined by the purpose of the engine, requirements to the size and order of operation of the cylinders located in one row. The disadvantage of this scheme is a more complex design of CRM and some inconvenience in maintenance, especially in the cramped conditions of the engine-boiler room of the ship. marine engines of this layout usually have from 6 to 24 cylinders and more.

Both two-stroke and four-stroke operating process is realised in trunk engines, but four-stroke trunk engines are predominantly used in the fleet. Relatively small mass of moving parts allows to force the engines of this type by speed without excessive increase of dynamic loads on bearings. In this connection, mainly medium and high-sided engines are built according to this scheme. At the same time, some Asian manufacturers, such as Akasaka Diesels, Hanshin EL, Makita, Matsui Iron Works, Niigata Engineering offer a range of low-speed trunk diesel engines. In particular, the Japanese company Hanshin EL produces six-cylinder diesels with cylinder diameters from 22 to 58 cm. The largest engine of this series, model LF58A, with a cylinder diameter of 580 mm and a piston stroke of 1050 mm develops power of 770 kW/cyl. at a speed of 190 min^{-1}.

times with a relatively small increase in its weight.

At present, double-acting engines (usually two-stroke) are not produced, as they are characterised by complexity of construction, very difficult working conditions of the piston group, rod and other parts. In such engines it is difficult to ensure good quality of gas exchange processes and especially mixture formation in the cylinder cavity through which the rod passes. The use of supercharging makes it possible to obtain the required power in simpler engines of simple action.

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CHAPTER 3

The design of marine internal combustion engines, the mechanisms and systems included in them. Forces and moments acting in marine engines

From a structural point of view, an internal combustion engine is a set of separate mechanisms and systems, each of which performs one or more functions. Given this, the division of the engine into separate mechanisms and systems is very conventional.

The crank mechanism includes the *cylinder-piston group* (*CPG*) (cylinder sleeve, piston assembly, cylinder cover) which absorbs the pressure of gases on the piston and the *motion group* which converts the reciprocating motion of the pistons into rotary motion of the crankshaft (basic parts, crankshaft, connecting rod, crosshead, piston rod, etc.).

The air-gas system supplies pre-cleaned air to the engine cylinders under atmospheric or increased pressure with pre-cooling of compressed air, and also removes combustion products from the engine, utilizes their heat and reduces exhaust noise. It includes (depending on the engine design) a *gas distribution mechanism*, a *system for purifying and supplying air to the engine (supercharging system)*, an *exhaust gas removal and exhaust noise reduction system, and an emission control system.*

The gas distribution mechanism is used to control the gas exchange processes of the slave cylinder in accordance with the set timing phases.

The engine air purification and supply system is used to take air from the ship's engine room or directly from the atmosphere, purify it from mechanical impurities and supply air to the engine cylinders. In most modern marine engines, this system includes a charge *air supercharging and cooling system* to increase the charge density at the engine inlet.

The *exhaust and exhaust noise reduction system* diverts combustion products from the engine into the environment.

The emission control system serves to reduce the emission of harmful substances in the engine exhaust gases.

It should be noted that engine mechanisms can combine several functions. For example, the pistons in the crank mechanism can partially or fully control the gas flows.

A fuel system that provides for the preparation and transportation of fuel and its subsequent injection into the engine combustion chamber under high pressure in a highly regulated manner.

The lubrication system is used to supply lubricating oil to all rubbing surfaces to reduce friction forces, cool rubbing parts and remove wear products from the friction zone.

Cooling system, serves to remove excess heat from the most heated in the process of engine operation parts, as well as from the working fluids.

Starting and reversing system, used to start the engine and change its direction of rotation.

Automatic regulation and control system, serves to maintain the specified mode of engine operation, to control the parameters of the operating process and the state of the main elements of the engine structure.

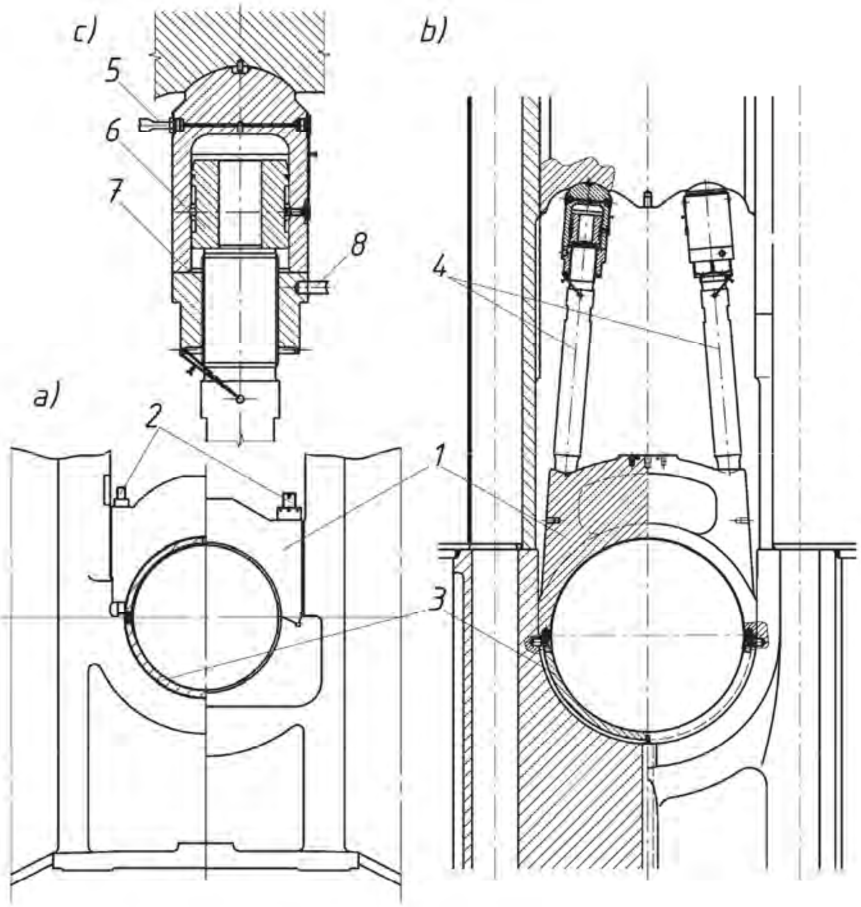
3.1 Fixed structural elements of marine engines

Regardless of design and purpose, the crank mechanism of any engine consists of fixed parts (backbone parts) and motion parts.

The fixed parts include: the foundation frame, the frame (crankcase), cylinder jackets, liners and their covers connected to each other. Together these parts make up the engine frame. Fixed elements of the crank mechanism of a four-stroke engine with a one-piece block-carter are shown in Fig. 3.1.

gines is done with hydraulic devices, which allows normalizing the tightening force.

Fastening of the frame bearing top cover with spacer hydraulic screw jacks is shown in Fig. 3.4 *b*. When mounting the cover, pressurized oil is supplied from the hydraulic power plant through connector *5* (Fig. 3.4 *c*) into the bodies of both jacks, which presses on the piston *6*. As a result, the threaded nut *7* is released, which is tightened with the help of a special rod *8,* after which the pressure in the jack is released and the position of the nut is fixed with a wire.



3.4 – Fastening of the upper cover of the frame bearing. *a* – by means of studs (MAN diesel engines of S, L and K series); *b* – by means of hydraulic screw jacks (Sulzer diesel engines of RTA series); *c* – device of hydraulic screw jack. 1 – bearing cover; 2 – studs; 3 – bearings; 4 – hydraulic screw jacks; 5 – oil inlet fitting; 6 – piston; 7 – compression nut; 8 – rod for cranking the nut (adapted from [7, 8])

3.1.2 Crankcase or crankcase box

The crankcase or crankcase box, depending on the engine design and its dimensions, is cast from steel or cast iron. For modern low-speed engines, the bed is made in the form of welded box structures made of sheet steel (Fig. 3.5). Boxed beds have high rigidity and a smaller number of bolted connections, which provides a good tightness of the crankcase. With a large length of the engine bed is made of two or three parts. In the lower part of the bed there is a buttress surface, which the bed rests on the foundation frame and is connected to it by bolted joints.

The bed is used to support the cylinder block. It connects the cylinder block to the foundation frame and forms an enclosed cavity for the crank mechanism, called the crankcase. In addition, the bed is the base member for the bulk of the engine auxiliaries, such as camshafts, fuel pumps, and valve train.

In order to prevent the destruction of the diesel engine on the side plates or hatch covers of the bed are installed spring-loaded safety valves 1 (Fig. 3.5), which are triggered automatically in case of explosions in the crankcase of the engine (Fig. 3.6).

Explosions may be caused by accumulated oil vapors, for the removal of which a crankcase ventilation system must be provided. The ventilation pipe, equipped with an oil separator and flame arrester, is usually routed from the crankcase cavity to the upper deck.

Vertical cast iron or steel plates reinforced on both sides with stiffening ribs, called parallels, are placed on the sides of the cross-braces of the bed to absorb the forces from the crosshead sliders.

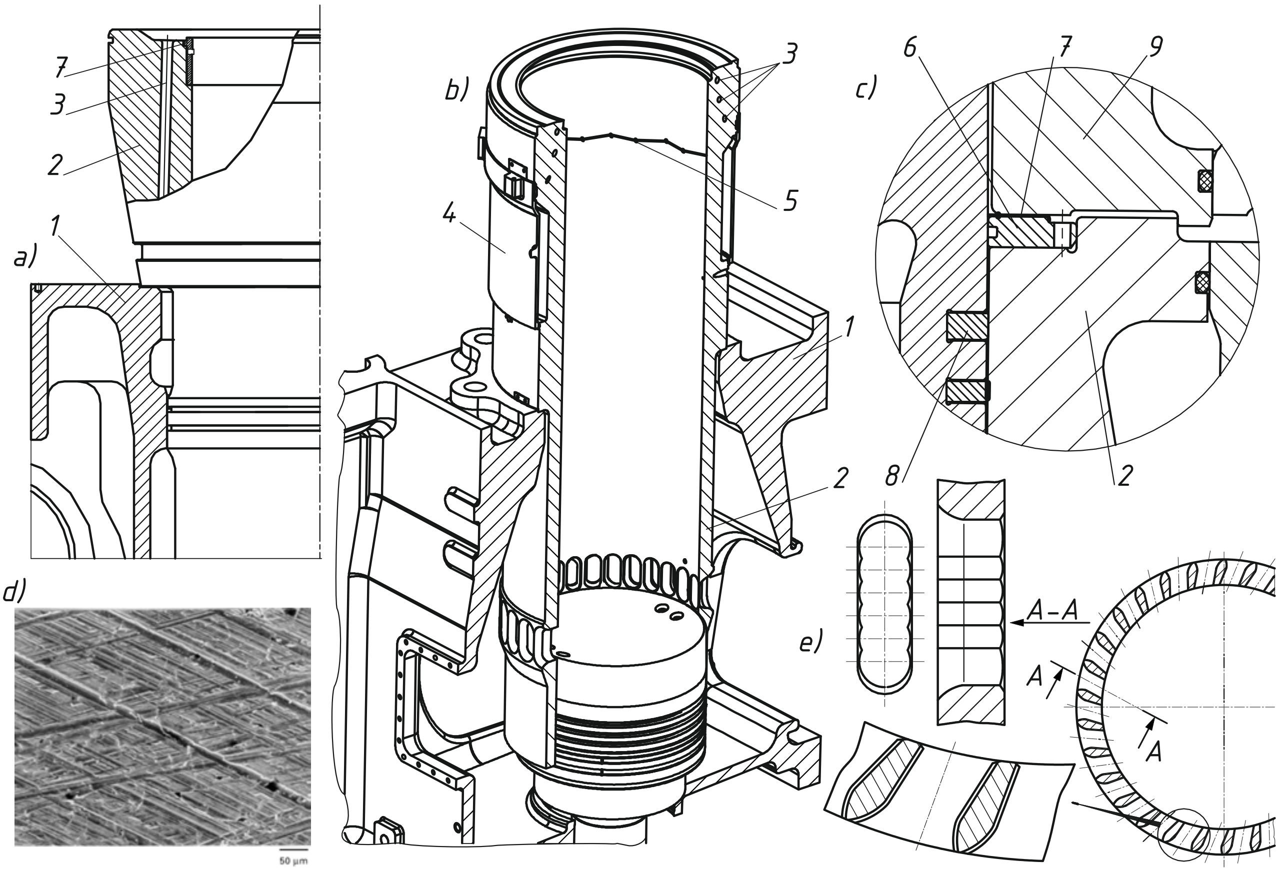


Figure 3.9 – Design and installation features of modern marine engines working cylinders: *a* – installation of cylinder sleeve of Wärtsilä L46 engine; *b* – installation of spacer between cylinder block and sleeve (Wärtsilä RT-flex 82 C series engines); *c* – installation of anti-polishing ring in MAN low-speed engines; *d* – microrelief on the working surface of cylinder liner; *e* – cross-section and direction of blow-through ports of Wärtsilä RTA series engines. 1 – cylinder block; 2 – cylinder sleeve; 3 – coolant channel; 4 – cooling jacket; 5 – oil distribution grooves; 6 – anti-polishing ring; 7 –steel gasket; 8 – piston rings, 9 – cylinder cover (adapted from [16, 17, 18, 19])

In some designs of high and medium speed engines, to reduce the height, part of the bushing is placed in the crankcase cavity, with slots made in the bushing to allow the connecting rod to pass through. Special slots are sometimes made in the upper part of the bushing to open the timing valves. When these design features are present, the bushing must be positioned in a specific position, which is achieved by means of locating pins.

A significant influence on the wear of the working surface of the sleeve is caused by carbon deposits formed on the cylindrical surface of the piston above the upper piston ring. These deposits contain mineral inclusions of high hardness and have an abrasive effect. To reduce wear of the sleeve surface on all modern engines, an anti-polishing ring is installed in the upper part of the sleeve (Fig. 3.9 *a*, *c*). It is a cylindrical insert in the upper part of the liner, which has an inner diameter smaller than the sleeve, but larger than the cylindrical part of the piston bottom. Thus, this ring forms a ledge at the level of the upper piston ring, which prevents the deposit of carbon deposits on the side walls of the piston head. In some engines, this insert also acts as a heat shield and is made of heat-resistant steel.

On two-stroke engines, the timing ports are cut into the lower part of the hub. In this place, to increase the cross-section of the bridges between the ports, a thickening is usually made, especially for engines with slotted purge schemes, where the exhaust port bridges are subjected to increased mechanical and thermal loads. In engines with straight-through purging schemes, the thickening of the belt in the area of the purge ports is not made, and the ports themselves are cut at an inclination to the axis of the cylinder sleeve to ensure the rotary movement of purge air in the cylinder (Fig. 3.9

speed engine. In case of low-power high-speed engines, aluminium alloys are sometimes used for manufacturing of covers. Structurally, cylinder covers are made both individually for each cylinder and for a group of cylinders, usually assembled in one block. In this case, the cover is called a cylinder block cover. This design is characteristic mainly for small engines. Cylinder caps of modern small-speed engines are made mainly of steel forgings with subsequent machining. The general view of the cylinder cover of a low-speed engine with valve-slotted blow-by is shown in Fig. 3.12 *b*, *c*.

Cylinder covers of two-stroke engines with straight-through valve purge and especially covers of four-stroke engines are quite complex products. Since with relatively small dimensions in them must be placed inlet and exhaust valves, channels for air inlet and exhaust gas, injectors, starting and safety valve, as well as indicator valve (Fig. 3.12 *e*, 3.13). In addition, the surface of the cover facing the working cylinder cavity, called the fire bottom, must be intensively cooled, for which purpose water circulation cavities are provided inside the cover. In addition, such a saturated structure must have a sufficiently high mechanical strength, e.g. in the low-speed MAN K98MC engine the pressure force of the gases on the cover can reach values equivalent to 1,100 tons. The temperature of the gases on the side of the fire bottom can exceed 1800°C, which explains the need for its intensive cooling.

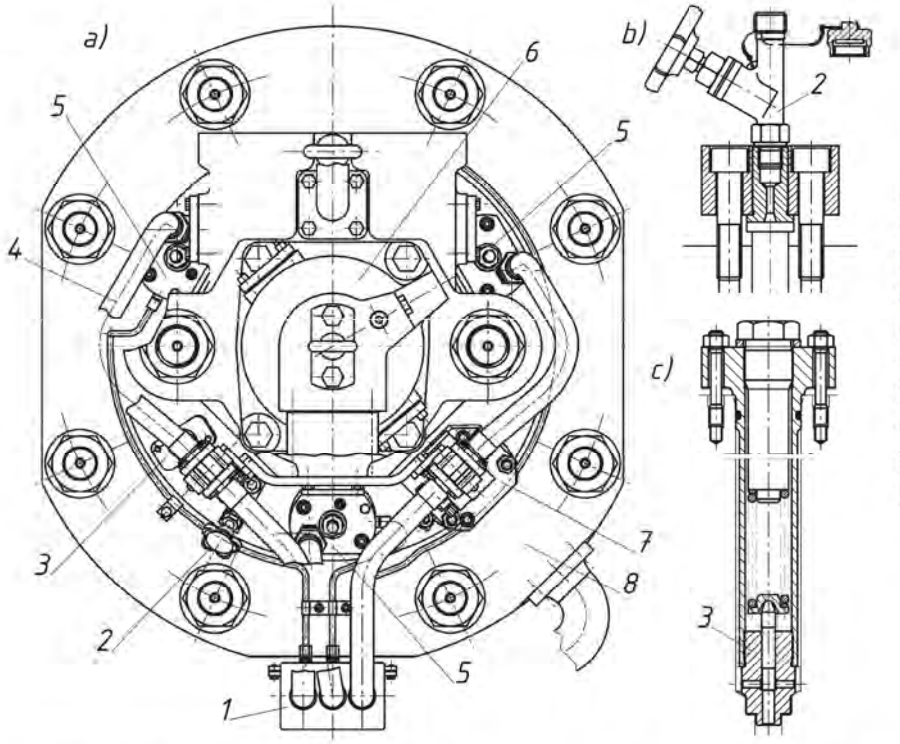


Figure 3.13 – Equipment of the cylinder cover of the two-stroke diesel engine RTA 72UB (*a*); indicator tap (*b*); relief valve (*c*). 1 – fuel distributor for injectors; 2 – indicator tap; 3 – relief valve; 4 – high pressure pipe; 5 – injection valve; 6 – exhaust valve drive; 7 – air start valve; 8 – cover flange (adapted from [18])

The cover is cooled with water coming from the space of the cylinder cooling jacket through special transition pipes or drillings in the cylinder liner flange, and is discharged from the highest point of the cover to avoid the formation of steam and gas locks. To more intensively remove heat from the fire bottom, two-level cooling is sometimes used in the covers of four-stroke engines. In this case, a second power bottom is placed above the relatively thin fire bottom, which is connected to the fire bottom through racks formed by wells of gas distribution valves, injectors, etc. (Fig. 3.12 *f*). The main fluid flow moves along the rear side of the firing bottom. In the case of high-force medium speed engine and high speed engine to increase rigidity, the fire bottom is made thick enough, and temperature stresses are relieved by passing water through a system of drilled holes in the bottom, as close as possible to the firing surface (Fig. 2.13 *f*). In order to prevent overheating of the exhaust valve, medium and high-speed engines operating on heavy fuels usually have a separate circuit for supplying coolant to the grooves in the valve seats.

To clean the cooling cavities, the cover is provided with hatches closed with covers or plugs. In

extended studs or bolts. At Caterpillar high speed engines some spread are composite pistons, in which the trunk part is connected with the bottom through the piston pin, which is simultaneously inserted into the holes on the trunk and on the bosses, made as a continuation of the head (Fig. 3.18).

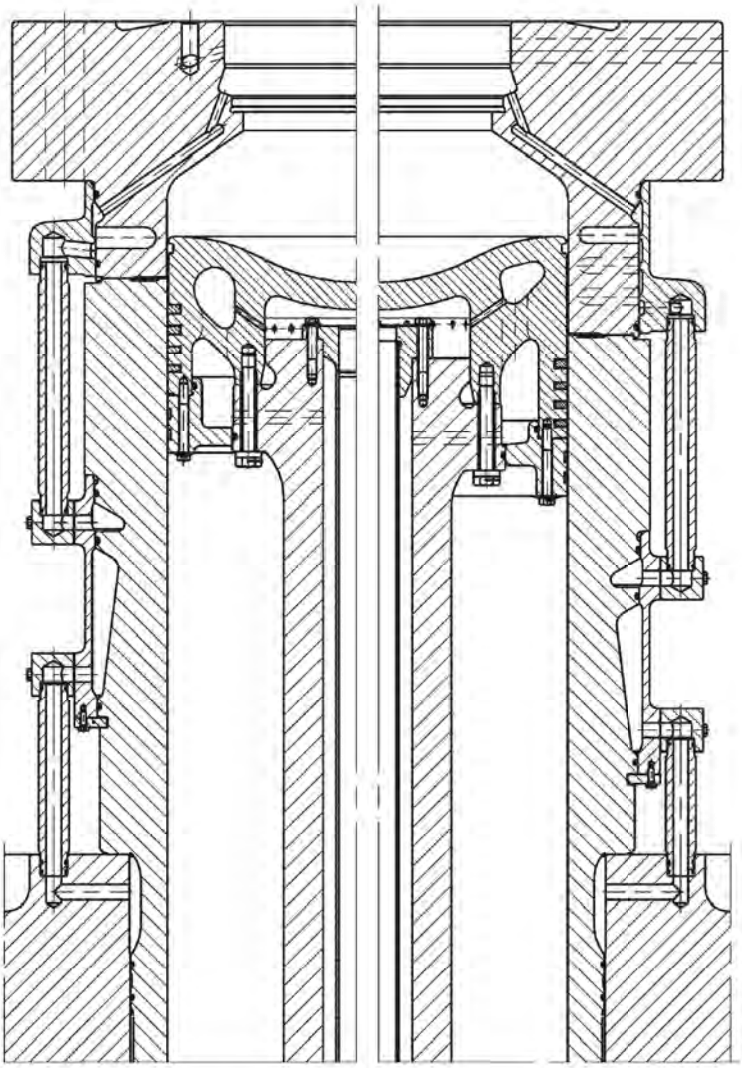


Figure 3.15 – Options for the combustion chamber of MC and MC-C engines with a low (left) and high (right) piston head (adapted from [28])

To strengthen the piston bottom, stiffening ribs are placed on it (Fig. 3.16 *c*), and the unsupported part is given a vaulted shape with high bearing capacity. Under the fire bottom of the piston in some designs is a second bottom, which closes the cooling cavity and acts as a thermal shield protecting the upper connecting rod bearing from excessive heating (Fig. 3.16 *a*, *b*, *f*).

Cooling of piston heads of modern high-force trunk engines is carried out by oil supplied from the general lubrication system. For this purpose, cavities are made in the head, which have the form of annular cavities, as well as radial or inclined drills 7 (Fig. 3.16 *b*, *c*, *d*, *f*). For intensive cooling of piston rings, these cavities are located above the caps.

Oil is supplied to the piston cooling cavity through a spring-loaded fitting 10 from the upper head of the connecting rod or through a drill hole in the piston pin, where it enters from the engine lubrication system through a drill hole in the connecting rod stem (Fig. 3. 16 *f*). Oil can be supplied in a continuous flow, providing flow cooling, or in individual portions. In the second case, the cooling cavity is not completely filled, and cooling occurs by agitating the oil when the piston changes direction.

From the cooling cavity, oil is drained into the lateral recesses on the outer side of the piston near the bosses, which are called coolers. In addition, a part of oil through the system of calibrated channels 4 is fed directly to the cylinder liner, directly under the oil rings.

loads during piston shifting, increasing the life of the cylinder. The use of a ball bearing allows the piston to be more accurately centered in the sleeve and the specific pressures to be more evenly distributed across the sleeve surface.

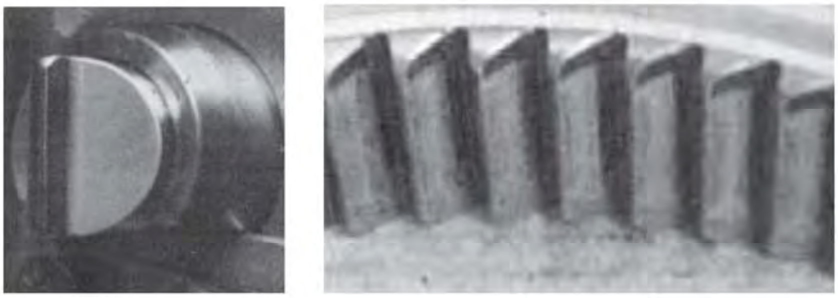


Figure 3.23 – Pawl (left) and fragment of the ratchet wheel of the piston rotation mechanism (adapted from [33])

3.2.3 Piston rings

For normal operation, the piston cavity must be sufficiently sealed and, at the same time, the piston must be movable relative to the cylinder sleeve. To fulfill these rather contradictory requirements, pistons are fitted with sealing elements called compression rings (Fig. 3.24).

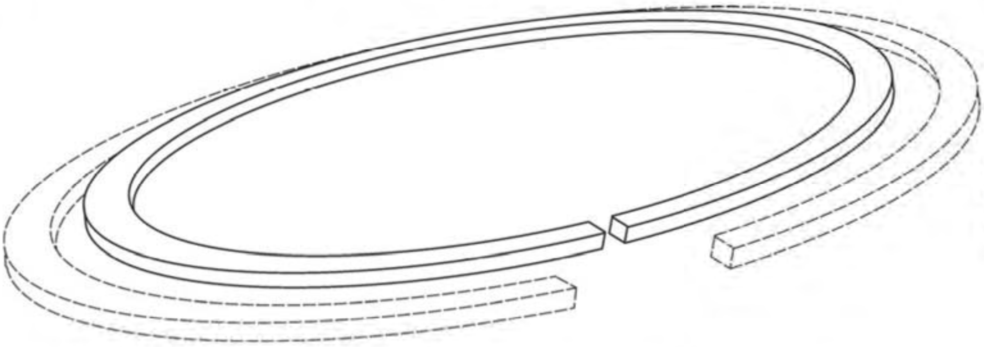


Figure 3.24 – Piston compression ring. The dotted line shows the ring in a free state

The number of compression rings depends on the engine design and usually ranges from 2 to 6. On modern engines, the number of rings and their height are tended to be reduced to reduce friction forces in the CPG. Structurally, the ring is a cut elastic element, which in radial section can have one of the shapes shown In Fig. 3.25 (*a-f*).

In the free state all piston rings are ellipse with a cut-out segment called a lock. When installed in the sleeve, the ring shrinks to a round shape and fits tightly to the surface of the sleeve (Fig. 3.24). The gap in the lock after its installation in the bushing should be sufficient to compensate for the thermal expansion of the ring during heating. Therefore, the value of the setting gap in the lock depends on the cylinder diameter and the maximum possible temperature of the ring.

Lock gaps are made straight (Fig. 3.26 *c*) or profiled (Fig. 3.26 *a*, *b*). On modern low speed engines the lock of the upper ring is made in the form of a complex profile shown in Fig. 3.26 *a*. Such locks provide a good seal of the gas joint, but are difficult to manufacture. All subsequent rings on the low speed engine are made with an oblique cut (Fig. 3.26 *b*), directing the slope of the joint in different directions. On four-stroke engine rings, the locking slot is made straight (Fig. 3.26 *c*). This lock shape provides less sealing efficiency, but is characterized by simplicity and reliability. The end gap between the ring and the cap must be sufficient to prevent the ring from jamming when the piston deforms under mechanical and thermal loads.

The rings are pressed against the working surface of the cylinder sleeve by their own elasticity. In this case, the maximum pressure is in the lock area. In order to reduce wear, various forms of pressure distribution patterns around the circumference are used. Particularly on low-speed engines, rings are used with the ends slightly bent inward. This correction serves to prevent the rings from hitting and breaking on the edges of the purge and exhaust ports.

In addition to the elastic forces, a significant role in pressing the rings, especially the upper rings, is played by the pressure of the gases, which acts on the back side of the ring.

At the point of rod passage through the diaphragm separating the cylinder piston cavity from the crankcase, an oil seal is installed (Fig. 3.32) preventing lubricating oil from the crankcase to the purge air cavity and purge air to the crankcase. The housing of the gland is made in the form of a flange consisting of two parts connected by bolts. On the inner side of the flange there is a number of annular grooves. In the upper groove is installed oil ring, which prevents the ingress of sludge from the piston cavity to the other rings. On the crankcase side, in the lower grooves, oil rings are installed to remove lubricating oil from the rod. From the lower grooves, the oil removed from the rod by the rings is returned to the crankcase through holes in the stuffing box housing. O-rings are located between the oil rings to prevent blow-by air from penetrating along the piston rod. Structurally, the O-rings and O-rings are a set of three or four segments that are held in place by locating pins. The segments are pressed against the piston rod by means of coupling springs located in grooves cut on the outer surface of the ring. On the inner surface of the oil sealing rings there are oil sealing lips. The sealing rings have a flat working surface. In one groove, the segments are placed in two rows with the locks offset relative to each other. The gaps between the ends of the segments ensure their fit to the piston rod as the segment wears.

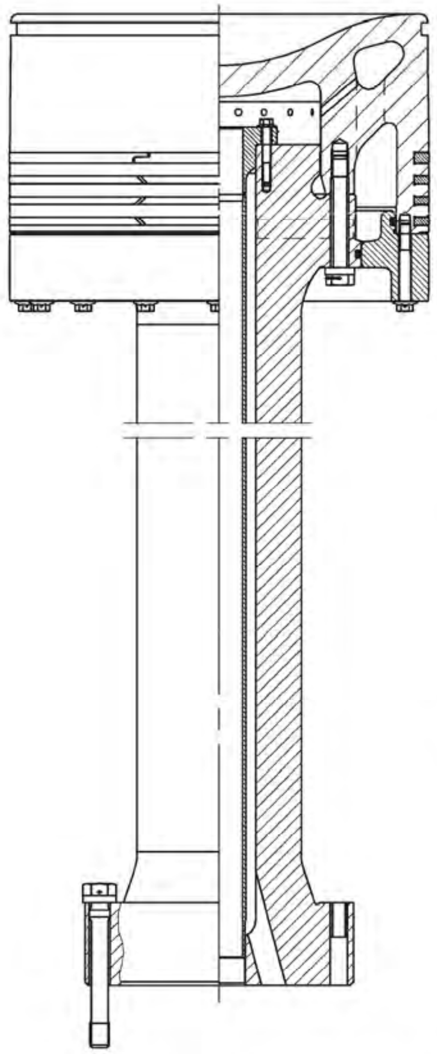


Figure 3.31 – MAN low speed engine piston and piston rod (adapted from [25])

Through the holes in the housing and the drilling in the diaphragm, the groove of the upper oil ring communicates with a funnel outside the engine, which is used to check the condition of the o-rings and oil rings. A burst of air indicates defects in the o-rings, and excess oil indicates defects in the oil rings. The oil seal is removed with the piston rod during inspection, but can also be disassembled in the crankcase without removing the piston.

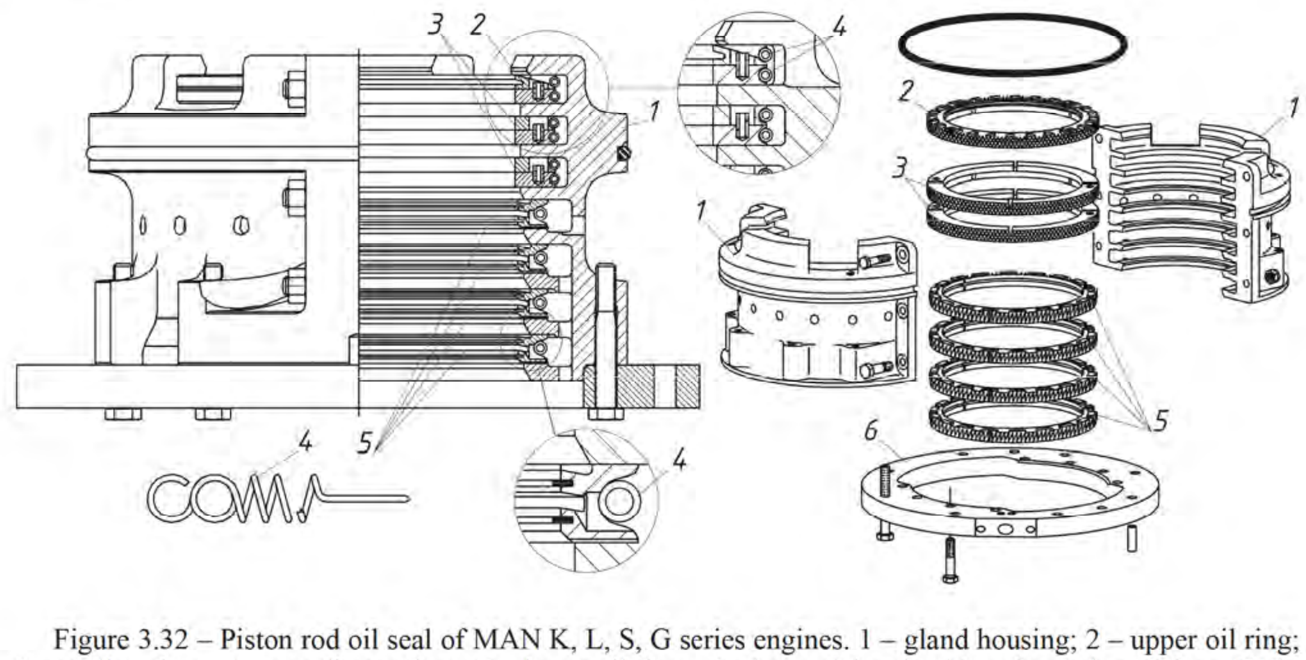


Figure 3.32 – Piston rod oil seal of MAN K, L, S, G series engines. 1 – gland housing; 2 – upper oil ring; 3 – sealing rings; 4 – coupling springs; 5 – lower oil rings; 6 – base of the gland housing (adapted from [21])

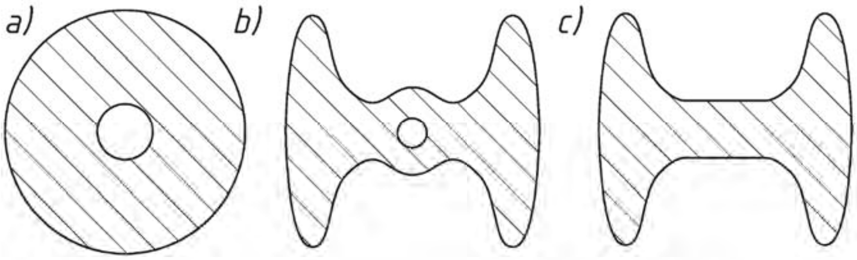


Figure 3.36 – Types of main sections of the connecting rod of marine engines: *a* – round with drilling for supplying oil to the upper head; *b* – I-beam with drilling; *c* – I-beam without drilling

The upper head of the connecting rod of a modern crosshead engine is made in the form of a split eye, which is a smooth continuation of the rod. The eye has a bore for the cross member bearing. The top cover has a cutout for the passage of the piston rod. To fix the position of the cover relative to the head housing, locating pins are used. The cross member bearing is made in the form of two liners, filled with a layer of white metal, and the lower liner is also covered with a running-in layer. For a more uniform supply of oil to the entire surface of the bearing, the liners have oil distribution grooves and thinning areas called pockets. They allow you to increase oil flow through the bearing, thereby improving the conditions for cooling the liners, as well as for removing wear products from the friction zone (Fig. 3.37).

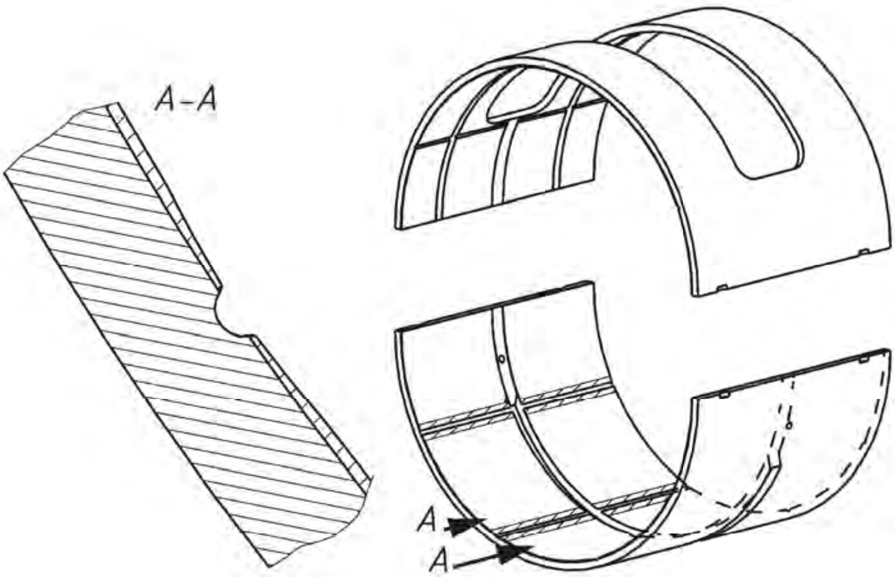


Figure 3.37 – Design of the LSE cross member bearing of the MC series from MAN (adapted from [21])

The upper head of the connecting rods MSE and HSE is usually made in the form of a solid closed eye, which has a smooth interface with the connecting rod and is symmetrical relative to its longitudinal axis. In some cases, the upper head is made with oblique cuts or profile recesses of the end surfaces (Fig. 3.35 *a*, *b*, *d*, *e*), which allows, without reducing the area of the most loaded lower part of the piston pin bearing, to reduce the total mass of the connecting rod. For engines with forced cooling of the pistons, the upper part of the head is made in the form of a cylindrical or spherical surface, to which a special spring-loaded cup is tightly pressed to supply oil from the connecting rod to the piston.

In a number of engines, the upper head can be made in the form of a spherical support (Fig. 3.35 *f*). This design allows you to more evenly distribute the loads over the surface of the bearing, and also, using a special mechanism located in the support, rotate the piston during operation (Fig. 3.16 *e*).

To facilitate piston removal, some engines have a removable upper head. The head is attached to the rod via a flange connection using bolts (Fig. 3.35 *e*).

To reduce friction, bearings in the form of bushings made of bronze or steel with an anti-friction layer are installed in the upper head. To ensure uniform distribution of oil, as well as supply oil to

gears or sprockets for driving auxiliary mechanisms, and a flange for attaching the flywheel on the parts protruding from the bearing (Fig. 3.43).

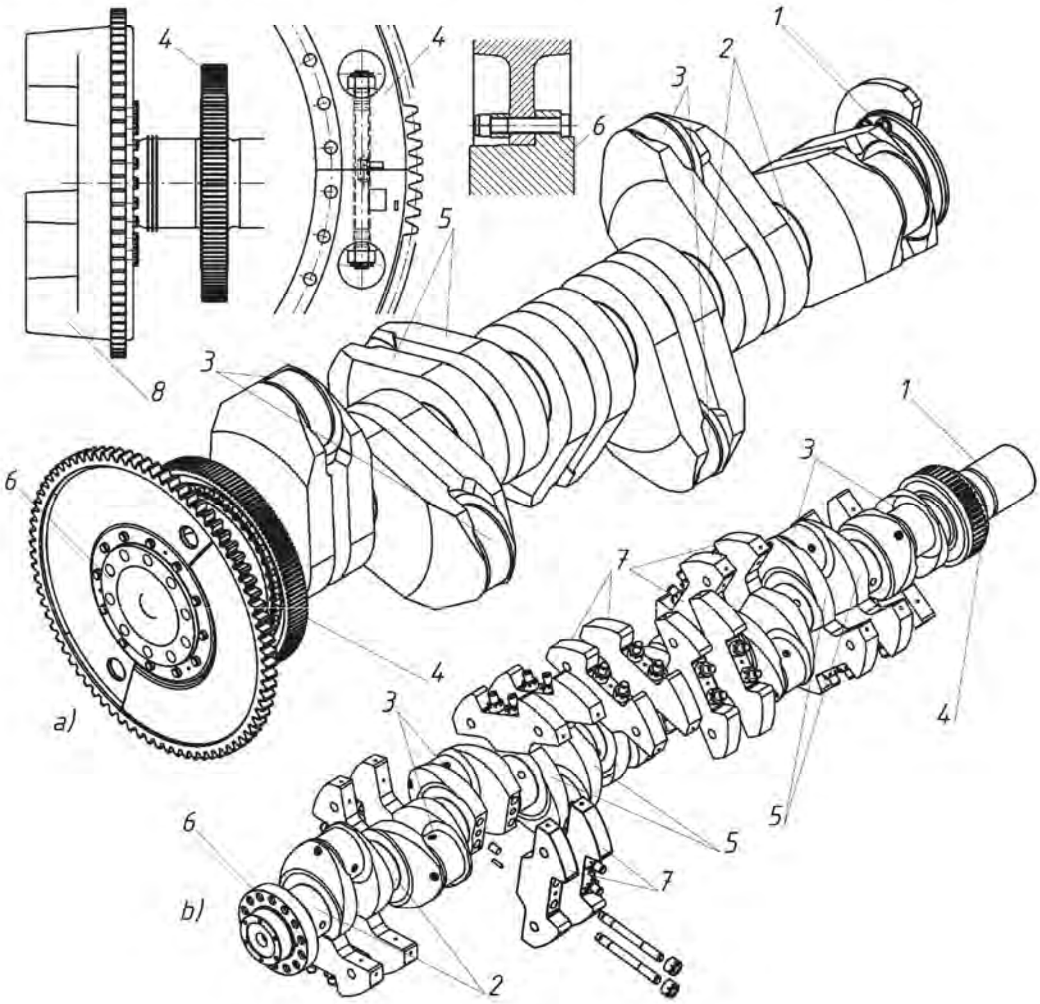


Figure 3.43 – Crankshafts of marine internal combustion engines. *a* – two-stroke low-speed diesel engine RT-flex60C from Wärtsilä; *b* – four-stroke diesel engine L21/31 from MAN; 1 – flange for driving auxiliary mechanisms; 2 – frame (root) neck; 3 – crank (crank) neck; 4 – sprocket (gear) for driving gas distribution and auxiliary mechanisms; 5 – crankshaft cheek; 6 – flange for mounting the flywheel; 7 – counterweights; 8 – flywheel (adapted from [44, 45])

LSE crankshafts with a number of cylinders from four to eight are made in the form of one section. With a larger number of cylinders, the shaft is made of two sections connected to each other using flanges. In this case, the sprockets or gears that drive the auxiliary mechanisms and the camshaft are placed in the central part of the engine.

Crankshafts of modern medium- and high-speed engines are usually made in one piece from a billet obtained by forging or casting.

The connecting rod journals in the longitudinal plane are displaced from the axis of rotation of the crankshaft by an amount equal to half the piston stroke, which is called the crank radius. In the plane of rotation, the connecting rod journals are offset relative to each other by an angle called the crank wedge angle φ_{wa}.

For two-stroke engines this angle is usually equal to:

$$\varphi_{wa} = 360/i,$$

for four-stroke:

$$\varphi_{wa} = 720/i,$$

where i is the number of engine cylinders.

multi-cylinder engines, auxiliary strokes are made due to working strokes in adjacent cylinders, so there is no need for a massive flywheel in such engines. In this case, only the gear for the turning device is mounted on the crankshaft.

The flywheel is usually attached to the rear flange of the crankshaft using bolts. The position of the flywheel is fixed by locating pins located asymmetrically to eliminate the possibility of incorrect installation. In addition, the pins relieve the flywheel mounting bolts from shearing forces.

A ring gear is located on the flywheel rim, which meshes with a shaft turning device used to rotate the engine shaft during maintenance and repair. For low-power HSEs, the flywheel crown is used during start-up to crank the engine using an electric or pneumatic starter.

The TDC marks of all cylinders and an angular scale for checking the valve timing and fuel supply are applied to the cylindrical surface of the flywheel (Fig. 3.47). In diesel generators, the flywheel is used to connect the engine to the generator coupling.

During operation, the crankshaft perceives forces of variable magnitude and direction resulting from gas pressure on the piston, inertial forces of reciprocating and rotationally moving masses, and moments created by these forces. Under the influence of all these factors, the shaft is subject to bending and torsion deformations. Periodic sign-alternating relative angular displacements of the crankshaft elements, occurring under the influence of time-variable and phase-shifted torques on individual cranks, are called *torsional*. The greatest danger to the crankshaft is resonant vibrations, that is, when forced vibrations of the shaft coincide with its own vibrations. The crankshaft rotation speeds at which resonance occurs are called *critical.*

In addition to torsional vibrations, under the influence of disturbing factors the shaft undergoes axial vibrations, which can disrupt the normal operation of the auxiliary mechanism drives.

To combat various types of vibrations of the crankshaft, special devices called dampers or dampers are installed on it. They eliminate these fluctuations or significantly reduce their amplitude.

3.3 Crankshaft bearings

3.3.1 Frame (main) bearings of the crankshaft

Crankshaft bearings are the most important engine components that influence its reliable operation and service life. According to their purpose, frame bearings are divided into support and thrust bearings. Support bearings limit the radial movements of the crankshaft, and thrust bearings limit the axial movements.

As support bearings for modern marine diesel engines, plain bearings are used, which are made in the form of steel liners filled with an antifriction layer of white metal or a tin-aluminium alloy.

In modern LSEs, the liners are installed in pairs, in the bore of the foundation frame and the cover (Fig. 3.4 *a*), or only in the bore on the engine foundation frame (Fig. 3.4 *b*).

Depending on the thickness of the steel base, bearings are divided into thick-walled and thin-walled.

Thick-walled liners have a steel base of sufficient rigidity (Fig. 3.48) to ensure stability of the geometric shape and maintain an anti-friction layer in areas where the liner does not have support, for example, in the area of the mating surfaces of the upper liner.

In some designs, for precise positioning of the liners relative to each other, during assembly they are tightened with tight-fitting bolts. To adjust the vertical gaps between the joints of the liners, spacers are installed.

Thin-walled bearing shells have a thickness of 2...2.5% of the journal diameter (Fig. 3.49). They do not have sufficient rigidity, so they must rest tightly along their entire length. For this purpose, bearings of this type are manufactured with a slightly increased circumference, due to which, when installing the liners and tightening the bearing cap, the necessary radial pressure is created between the liner and the bearing housing. The liners are secured against rotation by special cylindrical pins installed in a bore that overlaps the bearing housing and the liner.

Regardless of the design, an oil distribution groove is made on the inner surface of the upper

the influence of this deoxygenation is insignificant and in practical calculations formulas for the central crank mechanism are usually used.

Nowadays, disabling in excess of $e = 0.1r$ is used only in engines with special kinematic designs (for example, in engines with a rhombic mechanism or in engines with a parallel arrangement of each pair of cylinders operating on one crank.

Marine V-shaped engines use mechanisms with trailed connecting rods. A schematic diagram of a crank mechanism with a trailed connecting rod is shown in Fig. 3.53 c. One connecting rod in this mechanism is pivotally connected directly to the crankshaft journal and is called the main connecting rod, and the second is connected to the main connecting rod by means of a pin located on its head and is called the trailing rod (Fig. 3.35 *g*).

The system of the main and trailing connecting rods is used, for example, on 7FDS diesel engines from General Electric. In such a connecting rod system, high rigidity of the crank head of the main connecting rod is noted; however, the pistons connected to the main and trailing connecting rods have unequal strokes, since the axis of the crank head of the trailing connecting rod during operation describes an ellipse, the major semi-axis of which is greater than the radius of the crank (Fig. 3.53 *c*). The kinematics and dynamics of these mechanisms are covered in sufficient detail in the specialized literature; consideration of this issue is beyond the scope of this book.

When calculating for a central type crank mechanism (Fig. 3.53, *a*), the relationship between the piston movement S_x and the crankshaft rotation angle φ is determined as follows:

$$S_x = A'A = A'O - AO = (A'B' + B'O) - (AC + CO). \qquad (3.2)$$

The segment $A'B'$ is equal to the connecting rod length L_{cr}, and the segment $B'O$ is equal to the crank radius r. Taking this into account, as well as expressing the segments AC and CO through the product L_{cr} and r, respectively, by the cosines of angles β and φ, we obtain

$$S_x = (L_{cr} + r) - (L_{cr}\cos\beta + r\cos\varphi) = r\left[1 - \cos\varphi + \frac{\lambda_{cr}}{4}(1 - \cos\beta)\right]. \qquad (3.3)$$

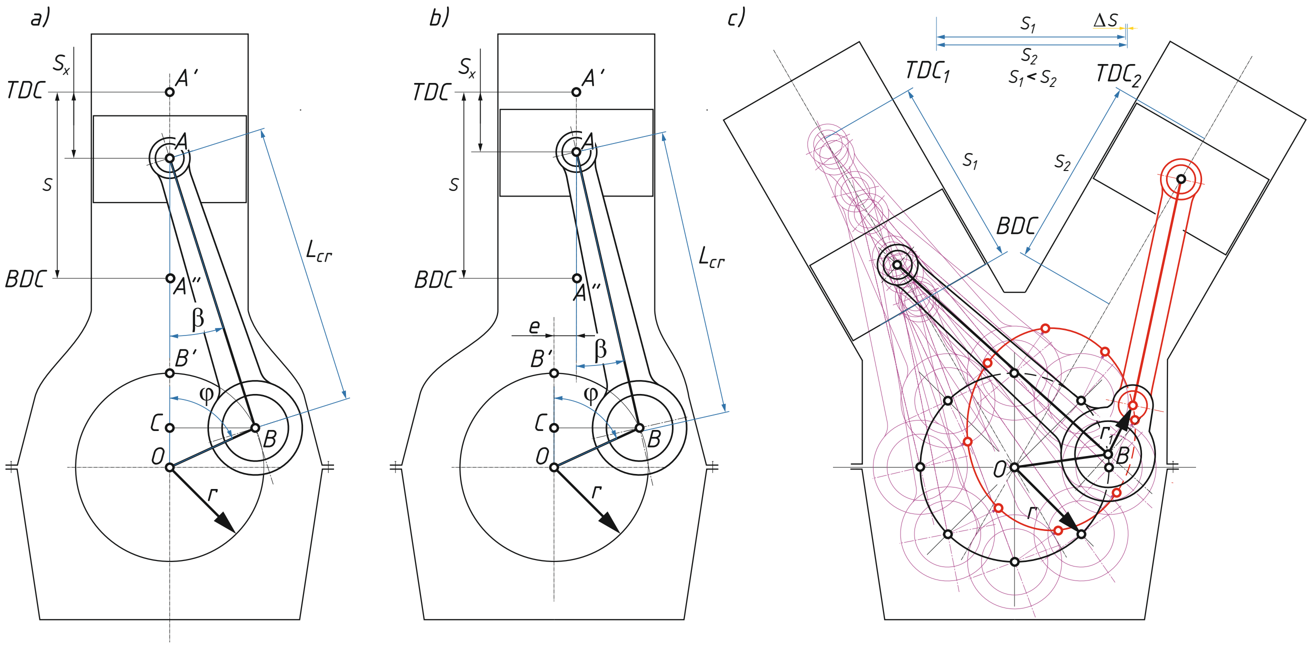


Figure 3.53 – Diagrams of crank mechanisms of internal combustion engines; *a* – central; *b* – displaced (disaxial); *c* – with trailed connecting rod

From the triangles ACB and OCB we find $CB = AB \sin\beta = OB \sin\varphi$ or $L_{cr} \sin\beta = r \sin\varphi$, from which:

3.4.2 Dynamics crank mechanism

When the engine is running, the following forces act on the crank of each individual cylinder:
- action of gases on the piston P_g;
- inertia of moving parts P_j;
- severity P_{gr};
- friction P_f;
- air resistance from the crankcase P_{atm}.

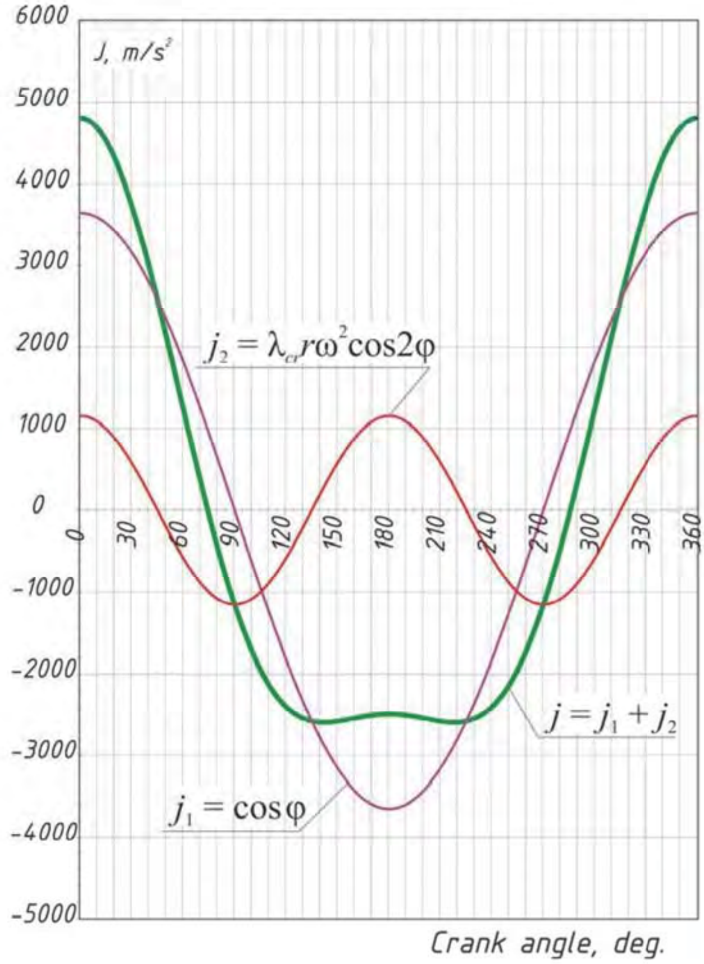


Figure 3.56 – Change in acceleration J of the piston depending on the angle φ of rotation of the crankshaft for a high-speed diesel engine

The forces P_{gr}, P_f, P_{atm} are relatively small, so they are usually neglected.

Forces from gas pressure P_g are associated with changes in pressure in the working cylinder, inertia forces P_j are determined by the masses of the moving parts of the crank mechanism, the piston stroke and the diesel engine speed.

Finding these forces is necessary to calculate the strength of diesel parts, assess the loads on bearings, determine the degree of uneven rotation of the engine crankshaft and calculate it for torsional vibrations.

When constructing a diagram of the forces acting in the crank mechanism, the initial force is the specific total force P_c acting on the pin (crosshead cross member). It represents the algebraic sum of the gas pressure forces P_g acting on the piston bottom, and the specific inertia forces P_j of the masses of parts moving back and forth,

$$P_s = P_g + P_j. \tag{3.14}$$

Let us consider in more detail the action of gas pressure forces on the piston P_g of inertia forces of moving masses P_j.

The force values from the gas pressure in the cylinder P_g are determined from the expression:

$$P_g = (p_i - p_0)\, F_p, \tag{3.15}$$

$$K' = S\cos(\varphi+\beta) = P\,(\cos(\varphi+\beta)/(\cos\beta)), \tag{3.19}$$

- tangential force T' tangent to the radius of the crank:

$$T' = S\,\sin(\varphi+\beta) = P\,(\sin(\varphi+\beta)/(\cos\beta)). \tag{3.20}$$

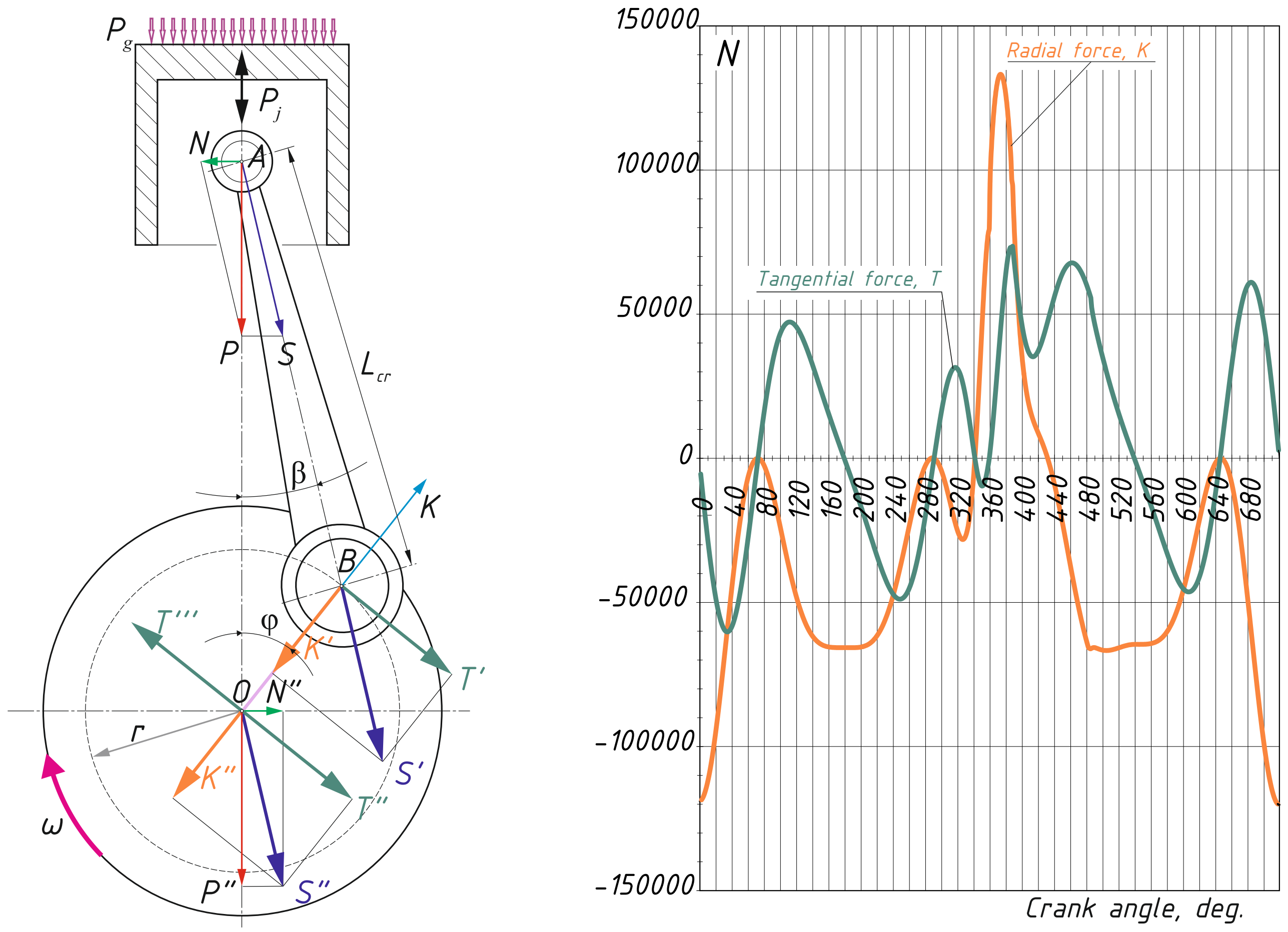


Figure 3.60 – Diagram of the decomposition of forces acting in the crank mechanism. Force decomposition S, directed along the connecting rod by force K directed along the radius of the crank and tangential force T

The normal force K' can be transferred along the line of its action to the center of the shaft and denoted as K'' ($K'' = K'$) (Fig. 3.60). To the center of the crank (point O) we apply two equal and oppositely directed forces T'' and T''' ($T' = T'' = -T'''$) then a pair of forces T' and T''' on shoulder r will create moment called torque M_{tor} (Fig. 3.61 *a*). The forces T'' and T''' balance each other. The physical meaning of these two forces is that the force T'' is applied to the support, which is the main (frame) bearing of the crank, and the force T'' is the reaction of the support to this force, as a result of which the crank axis O remains motionless.

The torque is transmitted through the crankshaft to the flywheel and is equal to:

$$M_{tor} = TR = Pr\,(\sin(\varphi+\beta)/(\cos\beta)) \approx Pr\,(\sin\varphi - (\lambda_{cr}/2)\sin 2\varphi). \tag{3.21}$$

Since the tangential force is variable (Fig. 3.60), the torque also changes from zero to maximum with a change in sign.

The forces K'' and T' can be added; their resultant S'', equal to the force S acting along the connecting rod, loads the main (frame) bearings of the shaft. The force S'' can also be decomposed into two components: N'', perpendicular to the cylinder axis, and P'', acting along the cylinder axis.

Forces N'' and N form a pair of forces (Fig. 3.61 *b*) on arm h, the moment of which is called overturning (M_{over}) and acts on the housing parts of the engine. The moment M_{over} is directed against the torque and, in accordance with the condition of equilibrium of the moving parts of the

within each cylinder is mutually balanced, therefore these forces are not transmitted beyond the engine frame. The cause of engine imbalance is the inertia forces of the reduced translationally moving masses P_j and the unbalanced rotating masses of the cranks of all cylinders, as well as the moments created by these forces.

Inertial forces are divided into inertial forces of translationally moving masses, which include the piston, rod and part of the connecting rod, and centrifugal forces created by rotating masses.

The inertial force of translationally moving masses is described by the formula:

$$P_j = m_j r \omega^2 (\cos\varphi + \lambda_{cr} \cos 2\varphi). \tag{3.24}$$

As has already been shown above, based on the complex nature of its occurrence, the inertial forces of translationally moving masses can be represented as the sum of two components:

$$P_j = P_{jI} + P_{jII},$$

where

$$P_{jI} = m_j r \omega^2 \cos\varphi = P_I \cos\varphi, \tag{3.25}$$

called first-order inertia forces, which obey the cosine law of the crank angle and

$$P_{jII} = m_j r \omega^2 \lambda_{cr} \cos 2\varphi = P_{II} \cos 2\varphi, \tag{3.26}$$

called second-order inertial forces, which obey the cosine law of twice the crank angle of rotation 2φ.

The forces P_{jI} and P_{jII} act in the direction of motion of the translationally moving masses along the cylinder axis and are applied to the axis of the crank head connection (Fig. 3.64). The values of the second-order inertia forces are significantly less than the values of the first-order forces, since expression (3.26) contains $\lambda_{cr} = 0.2...0.5$.

The external influence of inertial forces of the first and second orders of progressively moving masses is manifested in the fact that, being directed upward, they tend to tear the engine away from the foundation, and when directed downward, press it to the foundation. The frequency of the forces and the vibrations they cause will be a multiple of ω, respectively 2ω.

Inertia force of rotating masses K (3.23) at a steady rotation speed is constant in value, applied at the center of the connecting rod journal and directed along the radius of the crank from the center of rotation (Fig. 3.64). Depending on the position of the crank, the inertial force of the rotating masses tends to displace the engine from the foundation in planes that coincide at each moment with the plane of the crank and passing through the axis of the crankshaft. The force vector K, lying in the plane of the crank, will rotate, changing its position in accordance with the change in angle φ.

In the plane of the crank (plane V), the force K can be transferred along the radius of the crank to the axis of the crankshaft O the form of the radius vector as K'.

The force K', without disturbing its action, can be transferred parallel to itself to any point along the axis of the crankshaft by attaching to it a pair of forces K'' and K''' with a moment equal to the moment of force K' relative to point O_1 (Fig. 3.64 *a*). Using this rule, we bring the force K to the center O_s, which is the point of intersection of the crankshaft axis with the plane S in which the center of gravity of the engine lies. In this case, the force K''' is the reaction of the support to the force K''.

After reduction we have:

- force K applied to the center O and tending to tear the engine away from the foundation in the plane V;
- a pair of forces K' and K''', creating a moment M_k on arm l_1, acting in the same plane V and tending to tilt the engine relative to its center of gravity in a clockwise direction.

The resulting moment is called the moment of centrifugal force, it is equal to:

$$M_k = K l_1 = l_1 m_R r \omega^2. \tag{3.27}$$

The magnitude of this moment and the direction of its action depend both on the magnitude of the force K and on the position of the plane passing through the center of gravity of the engine relative to the cylinder axis.

In marine internal combustion engines, the centrifugal forces of inertia of the rotating masses, the inertial forces of the first and second order, as well as the moments caused by these forces, are balanced.

First-order inertia forces arise as a result of changes in the directions of movement of the piston group. These forces reach peak values when the piston moves at dead points. The consequence of the occurrence of first-order forces is transverse engine vibration, the frequency of which is equal to the crankshaft rotation speed. Typically these forces are partially balanced by balancers mounted on the crankshaft. Complete balancing of first-order inertial forces in this way is impossible, since the balancers themselves perform a rotational movement, and the balanced parts of the piston group are reciprocating.

Second-order inertia forces arise as a result of changes in the magnitude of the linear speed of movement of the piston in the process of moving it between dead points. These forces reach their maximum value in the middle of the piston stroke and cause transverse engine vibration, the frequency of which is twice the crankshaft speed. Second-order inertia forces are very difficult to balance, and since their magnitude is 2...4 times less than the first-order inertia forces, most often the second-order forces are left unbalanced so as not to complicate the engine design.

In addition to the external influence of inertial forces and their moments, the latter, acting inside the engine frame, load and deform its structural elements. Centrifugal forces, acting in the plane of the crankshaft, and inertial forces of the first and second orders in the plane of the cylinder axes, load the crankshaft and bearings and are transmitted to the foundation frame.

At the same time, the shaft is loaded by moments M_k, $M_{P_{j_1}}$ and $M_{P_{j_{II}}}$, which tend to bend the shaft in the planes of their action. The deformation of the shaft is perceived by the bearings (especially the central ones, which experience the greatest load) and the foundation frame in which they are located. As a result, bending and deformation stresses arise in the foundation frame, as well as on the shaft, under the influence of moments of inertia.

The noted action of inertial forces inside the engine frame determines the internal imbalance of the engine

It should be noted that regardless of the degree of external imbalance and the nature of its manifestation, the internal engine always remains unbalanced. The higher the rotational speed of the crankshaft and the greater the mass of the moving parts, the greater the inertial force and the greater the effect of their internal and external imbalance.

Balancing methods. External imbalance of the engine, especially if the unbalanced forces and moments are significant, leads to vibrations of both the engine itself and the ship's hull structures connected to it. Therefore, to reduce vibration, they resort to balancing the forces and moments occurring in the engine.

Balancing the centrifugal forces is carried out by installing counterweights on the crank cheeks (Fig. 3.65).

When rotating, the crank is acted upon by a centrifugal force K, lying in the plane of its rotation, which consists of two components: force K_1 from the mass of the connecting rod, attributed to the rotationally moving parts, force K_2 from the mass of the unbalanced part of the crankshaft.

The force K can be balanced by two counterweights mounted on the cheeks of the crank. They develop centrifugal force ($K^1_{cw} + K^2_{cw} \approx K$).

Then we can write:

$$m_R r\omega^2 = 2m_{cw} k\omega^2$$

or if the angular velocities of the engine masses and counterweights are equal

$$m_R r = 2\, m_{cw} k$$

where m_{cw} is the mass of one counterweight, kg; k – radius of inertia of the counterweight, equal to the distance from its center of gravity to the axis of rotation, m.

The centrifugal forces of inertia of rotating masses can be balanced in an engine with any number of cylinders by installing counterweights on the crankshaft. In most multi-cylinder engines, the

in one direction or the other.

If the rigid fastening of the free end of the shaft is replaced with significant resistance, for example, by installing a propeller or a generator flange at the end of the shaft, the loaded end of the shaft will rotate uniformly, and the free end will oscillate.

Reviewed in Fig. 3.69 and the torsional system is the simplest single-mass system. If another disk is attached to the free end of the shaft (Fig. 3.69 *b*), we obtain a two-mass torsional system. If to disks masses m_1 and m_2 apply moments M_{tor} and M_{res} of opposite signs, the shaft will spin in opposite directions. When the action of the moments ceases, elastic vibrations will occur relative to a certain section in which the angle of twist will be equal to zero. This section of the shaft is called the oscillation node. If on a scale diagram we plot the maximum values of the angular amplitudes of vibrations and connect the resulting points with a straight line, we will obtain a graph of the angular amplitude deviations of masses from the equilibrium position (Fig. 3.69 *b*). A two-mass system has only one oscillation node in the *g* plane, and is therefore called a single-node system. The actual torsional system of a ship's propulsion installation is a complex multi-mass system in which the number of oscillation nodes is less than the number of masses per unit. In practice, when determining the frequency of free oscillations, the propulsion installation circuit is simplified and leads to a three-mass system: engine-flywheel-propeller (Fig. 3.69 *c*). Such a system can have both single-node and two-node vibration modes, each of which has its own free vibration frequency. The more nodes in the system, the higher the frequency of its oscillations.

Forced oscillations in the shafting system arise under the influence of periodically changing torques from gas pressure forces in the cylinders and inertia forces of progressively moving masses. These moments are called disturbing moments.

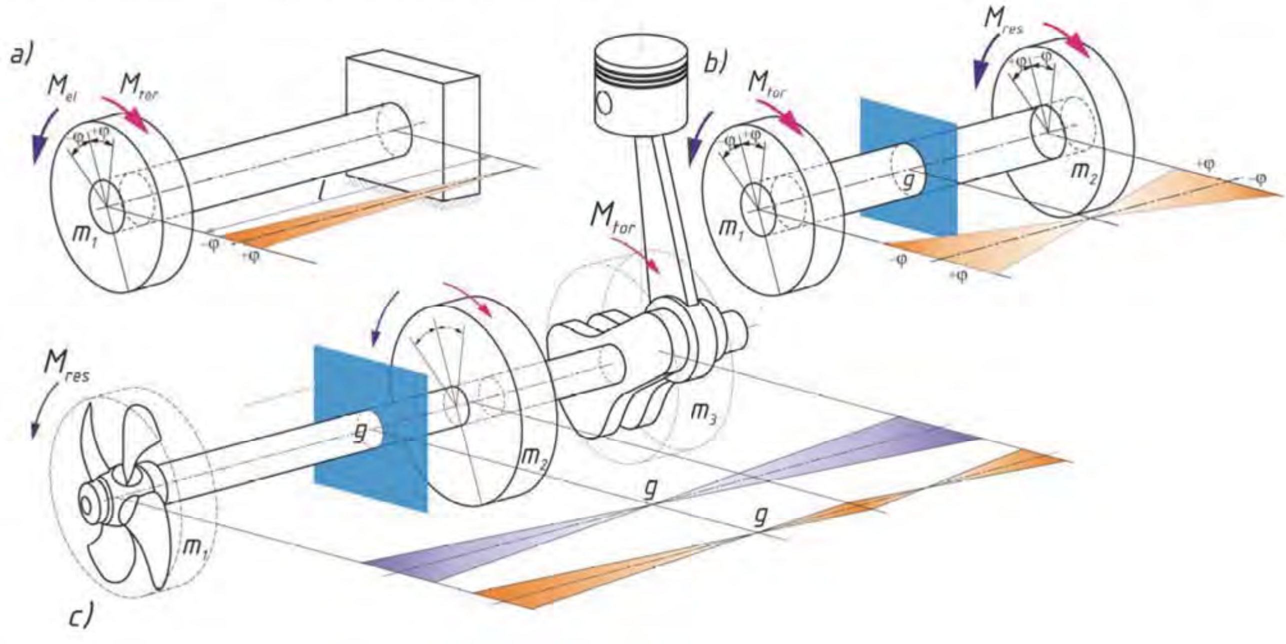


Figure 3.69 – Torsional systems: *a* – single-mass; *b* – two-mass; *c* – three-mass

Forced oscillations in the shafting system arise under the influence of periodically changing torques from gas pressure forces in the cylinders and inertia forces of progressively moving masses. These moments are called disturbing moments.

As has already been shown above, the forces P_{jI} and P_{jII} and the moments they create can be decomposed into elementary components – harmonics, varying with different amplitudes and frequencies. This decomposition is called harmonic analysis.

Each harmonic has its own order, which shows the number of its complete changes per one revolution of the shaft. Each harmonic excites forced oscillations in the shaft of a certain frequency, proportional to the order of the harmonic and the shaft rotation frequency. The largest amplitudes of forced torsional vibrations are caused by harmonics, the order of which is equal to or a multiple of

generated during damper operation, continuous circulation is provided – the oil is constantly drained through drainage holes in the chamber housing. The diameter of the holes is relatively small and they do not affect the operation of the device.

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SECTION 4

Air-gas systems of marine engines

Marine internal combustion engines, like most other piston and rotary vehicles for transport purposes, use air oxygen, which enters the engine from the surrounding atmosphere, to burn fuel in their working space. Interacting with the fuel, a mixture of unreacted air components with combustion gases is formed in the engine working space, which is used as a working fluid that performs useful work during the expansion process. In addition, atmospheric air is used to clean the working cylinders from combustion products. Effective cleaning of the working space with atmospheric air is especially important in two-stroke engines that do not have a displacement piston stroke.

An equally important process is the removal of engine exhaust gases into the environment, the efficiency of which largely determines the economic, environmental and a number of other important indicators of engines.

4.1 Use of atmospheric air in the operating process of marine engines

4.1.1 Atmospheric air is the most important component necessary for organizing the operating process in engines with an open thermodynamic cycle. It is used to form the working fluid at the compression and expansion stage, and at the combustion stage, the oxygen contained in the air is used as an oxidizer.

In addition to the direct use of atmospheric air as the main component for the formation of working fluids, the earth's atmosphere is used by marine engines as a natural refrigerator, into which a significant part of the process heat and fuel combustion products are removed.

Atmospheric air is a mixture of gases, consisting mainly of nitrogen and oxygen. Taken together, the share of these two components is 98...99% depending on the concentration of water vapor (humidity). In addition to the two named components, dry atmospheric air contains in small proportions: carbon dioxide, hydrogen, methane, inert gases and other elements (Table 4.1).

Table 4.1 – Chemical composition of dry air

Substance	Designation	By volume, %	By weight, %
Nitrogen	N_2	78.084	75.5
Oxygen	O_2	20.946	23.15
Carbon dioxide	CO_2	0.03	0.046
Hydrogen	H_2	0.0005	0.00008
Methane	CH_4	0.0002	0.000084
Noble gases	–	0.9365	1.2965
Other components	–	–	0.007323

A significant influence on the concentration of gases in the air is exerted by water vapor, the content of which depends on temperature, humidity, time of year, climate, etc. Thus, at a temperature, 0°C cubic meter of air can contain up to 5 gram water, and at a temperature + 10°C bout 10 gram.

When taken from the atmosphere, air is subjected to filtration, removal of excess moisture and dilution with crankcase gases or combustion products, for example, during exhaust gas recirculation in order to reduce oxygen concentration.

When calculating the engine operating process, the composition of the air entering it can be adjusted depending on the methods used to prepare the charge for supply to the cylinder, but by default it is generally accepted that the main components of the charge are nitrogen and oxygen. The influence of other components of the gas mixture on the operating process can be neglected due to

enization is required (uniform distribution of fuel molecules in the charge volume, when the reacting substances are in the same state of aggregation, namely gaseous). If this condition is not met, local zones are formed inside the charge, in which all the oxygen is consumed and further combustion is impossible, and the supply of oxygen from other zones, where it is in excess, physically takes up the time allotted for the combustion process. A high degree of homogenization is achieved in engines with external mixture formation, operating on light liquid (gasoline) or gaseous fuels. In such engines, the fuel to air ratio is maintained close to stoichiometric.

In diesel engines, obtaining homogeneous mixtures is difficult due to the peculiarities of mixture formation, in which fuel enters the combustion chamber in liquid form, forming a heterogeneous system at the first stage with the cylinder charge (when the reactants are in different phase states). During the process of heating and evaporation of the fuel, the system goes from a heterogeneous state to a homogeneous state, but this requires additional time. Limiting the time allotted for combustion significantly affects the completeness of fuel burnout and the efficiency of its use.

Excess air coefficient. To increase the reaction rates during internal mixture formation in diesel engines, an excess of oxidizer is provided in the volume of the combustion chamber, that is, the actual amount of supplied air L significantly exceeds the theoretically required L_0. The relationship between the actual and theoretically required amount of air for combustion is called the *excess air coefficient* and is denoted by – α,

$$\alpha = L/L_0.$$

For a given excess air ratio, the actual number of kilomoles for 1 kg fuel combustion will be:

$$L = \alpha L_0, \text{ kmol/kg}.$$

Air mass:

$$l = 28.95\ \alpha L_0, \text{ kg/kg}.$$

According to experimental data, for modern supercharged diesel engines α varies within the following limits (Table 4.2):

Table 4.2 – Excess air coefficient for various types of marine diesel engines

Low-speed diesel engines	Medium-speed diesels	High-speed diesels
2.5…3.5	2.0…3.2	1.8…2.6

The tendency to increase the excess air ratio in modern marine diesel engines is also due to the need to reduce the maximum cycle temperature, which leads to a decrease in the content of nitrogen oxides in the exhaust gases. Reducing the temperature is especially important for two-stroke engines, where the amount of heat transferred to the piston from the combustion products, other things being equal, is twice as much as in four-stroke engines, which increases their thermal intensity.

However, not all of the air supplied to the cylinders is directly involved in the operating process. Some of it, mixed with exhaust gases, goes into the exhaust system. Thus, the mass of air involved in combustion G_{air} will always be less than the mass of air supplied through the gas distribution elements G_s. This makes it possible to improve the cleaning of the cylinders from combustion products remaining from the previous cycle and ensure optimal temperature conditions for the piston group parts due to additional heat removal.

The ratio $\varphi_a = G_s / G_{air}$, characterizing the air consumption for purging and filling the cylinder and is called the *purge coefficient*. The higher its value, the greater the air consumption for gas exchange and the energy consumption for driving the charging units. In two-stroke low-speed diesel engines that do not have pump strokes φ_a =1.6...1.65, in four-stroke medium- and high-speed diesel engines – φ_a =1.0...1.2.

From the air receiver, a certain amount of air enters the cylinder through the intake valves (in four-stroke engines) or ports (in two-stroke engines). Part of the air (excess purge air) leaves the cylinder along with the combustion products. The rest, most of the air G_{air}, called *mass charge*, remains in the working cylinder at the end of the gas exchange.

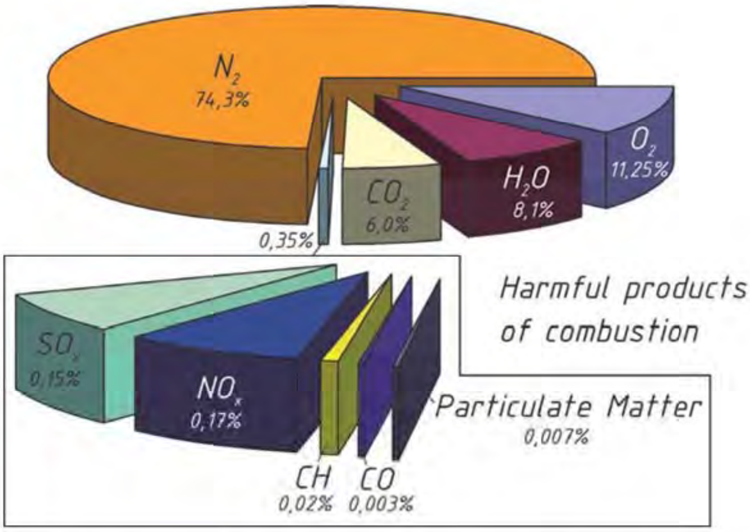

Figure 4.1 – Approximate composition of combustion products of a marine engine when using heavy fuel containing 4% sulfur (adapted from [1])

The amount of *complete combustion products* formed during 1 kg fuel combustion is determined by the previously given stoichiometric combustion equations (in kmol/kg of fuel):

$$M_{CO_2} = \frac{C}{12};\ M_{H_2O} = \frac{H}{2};\ M_{SO_2} = \frac{S}{32}.$$

During combustion with excess air (α > 1), the oxygen entering it does not participate in the reaction in full. Amount of excess oxygen O_2:

$$L_{O_2} = \alpha L_0 - L_0 = (\alpha - 1)L_0, \text{ kmol/kg fuel,}$$

hence:

$$M_{O_2} = 0{,}21(\alpha - 1)L_o, \text{ kmol/kg fuel.}$$

The remaining number of moles in the combustion products comes from inert nitrogen N_2, which is part of the air:

$$M_{N_2} = 0{,}79 L_{O_2} = 0{,}79\alpha L_0.$$

The total quantity M of products of complete combustion of one kilogram of fuel at α > 1 is determined by summing all components:

$$M = \frac{C}{12} + \frac{H}{2} + \frac{S}{32} + (\alpha - 0{,}21)L_0, \text{ kmol/kg fuel.}$$

From the law of conservation of mass, it follows that the mass of combustion products is equal to the sum of the masses of air and fuel before combustion, while the volumetric quantities of the working fluid before and after combustion are not equal. During chemical combustion reactions, the number of molecules of gaseous products increases, resulting in an increase in volume (number of kilomoles). This increase can be determined by the difference:

$$\Delta M = M - L = M - \alpha L_0.$$

To estimate the increase in volume (number of moles) of combustion products, *the theoretical (chemical) coefficient of molecular change is used* β_0, which represents the ratio of the number of moles of combustion gases M to the number of moles of air L without taking into account the residual gases in the cylinder:

$$\beta_0 = \frac{M}{L} = \frac{L + \Delta M}{L} = 1 + \frac{\Delta M}{L}.$$

In addition to the components listed above, the combustion products contain residual gases M_r,

- release of gases into a common manifold from all cylinders;
- release of gases into a separate pipe.

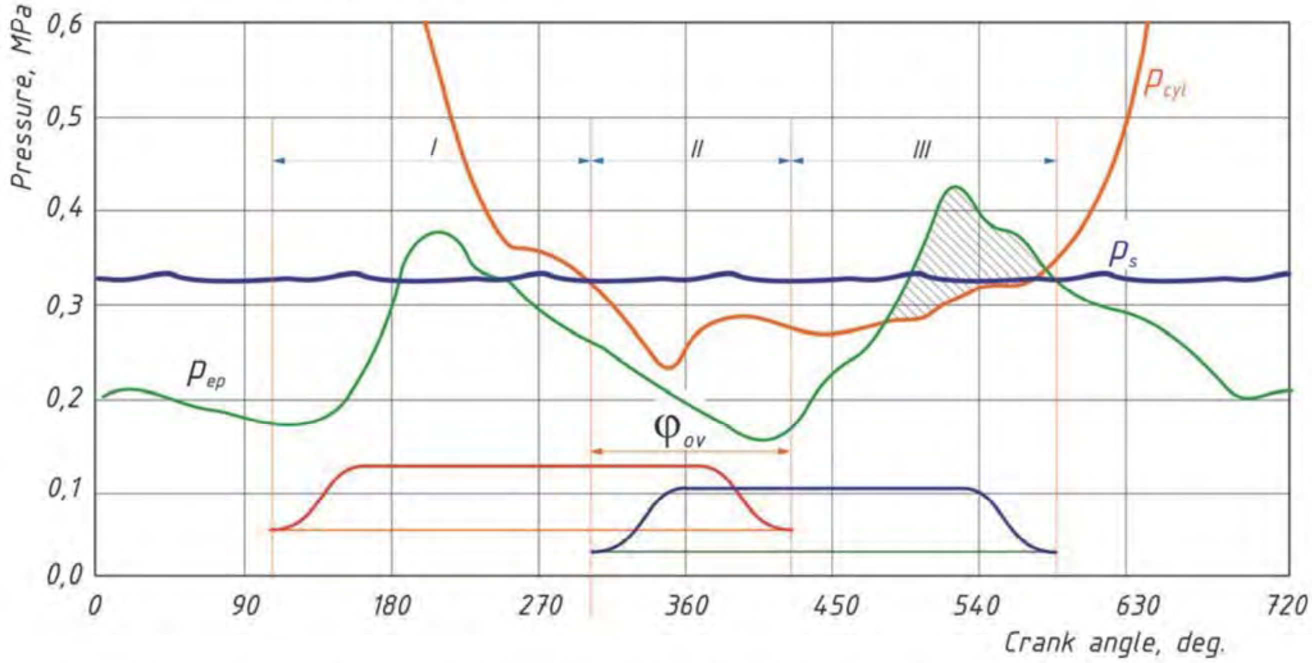

Figure 4.3 – Changes in pressure in the working cylinder p_{cyl}, inlet receiver p_s, exhaust manifold p_{ep} during the gas exchange process of a four-stroke engine combined with the opening profiles of the gas distribution valves (adapted from [2])

For successful purging of the combustion chamber, the valve overlap section (φ_{ov}, section *II*) is selected in such a way that at this interval the pressure in the intake receiver p_s is higher than the pressure in the exhaust manifold p_{ep}. The higher the boost pressure, the longer the valve overlap can be, the better the cleaning of the cylinder from combustion products can be done. Section *III* shows a case where an increase in pressure in the common exhaust manifold associated with exhaust in one cylinder leads to exhaust gases being thrown into the other cylinder (shaded area) through the open intake valve. Thus, fluctuations in pressure p_g (Fig. 4.3) from the exhaust of neighbouring cylinders can create back pressure at the exhaust and make it difficult to clean the cylinder. To avoid this harmful phenomenon, the exhaust from individual cylinders with similar operating orders is carried out into separate exhaust manifold pipes, which are combined into one pipe at a distance from the valve sufficient to reduce the influence of pressure pulses on neighbouring cylinders.

In *two-stroke engines* with direct-flow valve scavenging, the air flow moves along the cylinder axis with layer-by-layer displacement of combustion products through a valve installed in the cylinder cover. Since in this type of engine there are no piston pump strokes, the filling process is combined with the exhaust process. These processes occur only during part of the piston stroke, the magnitude of which is determined by the moment the exhaust valve closes (Fig. 4.4). All modern two-stroke low-speed engines have purge ports located in the lower part of the cylinder liner; the phases and laws of their opening and closing are determined by the kinematics of the piston movement. The opening and closing phases of the exhaust valve, the height of its lowering, and the law of motion are set by the profile of the camshaft cam or by an electronically controlled hydraulic actuator.

The gas exchange process in a two-stroke engine is conventionally divided into four phases (Fig. 4.4 *a*): free exhaust I; forced release II; purging and filling III, charge loss IV. In this case, phases II and III occur simultaneously, fresh air enters the cylinder through purge ports opened by the piston as it moves to BDC, and combustion products displaced by air are removed through an open exhaust valve located in the cylinder cover. The change in the flow sections of the gas exchange organs according to the angle of rotation of the crankshaft is called «area-time» diagrams

The efficiency of exhaust gas removal into the atmosphere means their release with minimal back pressure, the use of exhaust gas energy, both to obtain mechanical work and thermal energy, and changing the chemical composition of exhaust gases in order to reduce the content of harmful components in them.

The issue of effective gas exchange between the working cylinders of internal combustion engines and the external environment has become relevant almost from the moment of their creation, and remains so to the present day.

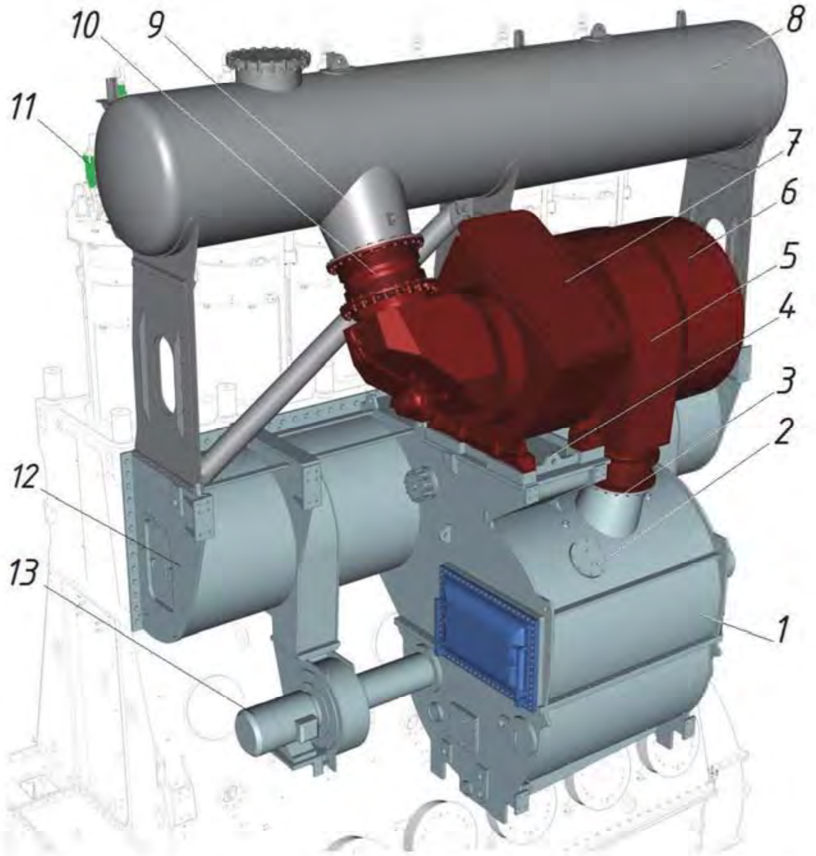

Figure 4.6 – General arrangement of elements of the air-gas system of a modern LSE from MAN: 1 – housing of charge air coolers and moisture separators; 2 – hatch for monitoring and cleaning the cooler; 3 – air duct; 4 – platform for turbochargers; 5 – compressor part of the turbocharger; 6 – air filter; 7 – turbocharger turbine; 8 – exhaust receiver; 9 – gas duct; 10 – thermal expansion compensator; 11 – exhaust valve assembly; 12 – air receiver; 13 – electric blower (adapted from [3])

Over the entire period of existence of piston engines, the mechanisms and assemblies of the air-gas path have gone through a certain development path from low-power engines with cylinder filling under atmospheric pressure to modern systems equipped with supercharging units that make it possible to increase engine power several times.

4.5 Gas distribution mechanisms of marine engines

4.5.1 History of the creation and development of gas distribution mechanisms for marine piston internal combustion engines

When developing the first piston engines, their creators relied heavily on the experience of designing piston steam engines, borrowing from there both ready-made technical solutions and individual components and assemblies. Thus, in the first piston internal combustion engine, which had some commercial success, its creator, the Belgian engineer Etienne Lenoir*, used a spool valve gas distribution system, borrowed from steam engines with only minor modifications (Fig. 4.7).

However, spool-type mechanisms with a sliding spool plate, with the help of which steam was distributed under pressure, turned out to be ineffective in internal combustion engines, in which the working space was filled under atmospheric pressure. The small flow area of the gas distribution organs, large friction losses, which increased significantly with increasing rotation speed, the complexity of the drive from the additional crankshaft journal and a number of other disadvantages forced us to very quickly abandon the use of such mechanisms in internal combustion engines.

Along with the improvement of valve mechanisms, at the inception of engine construction, many developers tried to improve spool valve timing. Thus, at the beginning of the twentieth century, along with the already mentioned flat spool mechanisms, the so-called sleeve gas distribution was considered promising. For the first time, such a scheme with sliding sleeves was developed by the American engineer Charles Yale Knight* in 1903, and is often called the «Knight system» after his name. Knight's engine used two concentric reciprocating moving sleeves. Each sleeve was equipped with large ports on one edge. As the liner was moved up and down, these cutouts periodically aligned with the inlet or outlet port in the side wall of the cylinder. The liners were driven by an intermediate eccentric shaft, which replaced the cam shaft, which rotated at half the speed of the crankshaft. This mechanism worked very well in engines with relatively low power, but when trying to obtain high litre power, engines with a double sleeve often failed due to oil starvation of the developed friction surfaces, and therefore they were subsequently abandoned. In England, work on engines with liner gas distribution was carried out by Gary Ricardo*, who spent thirty years conducting research on sleeve gas distribution. In the course of these and similar works, versions of engines were developed in which the liners rotated around the cylinder rather than sliding along it, engines with cylinders whose ports were located in the head of the block, and not in the side walls, etc. Thanks to the work carried out, many designs of sufficiently powerful gasoline piston engines for aviation needs were created, and only the advent of gas turbine engines led to the curtailment of these works.

The first two-stroke reversible marine engine was built by Sulzer in 1905. This was the engine in which a direct-flow valve timing scheme was first used, in which intake was carried out through valves located in the cylinder cover, and exhaust through exhaust ports cut into the lower part of the working cylinder sleeve. Having a cylinder diameter 175 mm and piston stroke 250 mm, in a four-cylinder version the engine developed a power of 74 kW at 375 min^{-1} (Fig. 4.11).

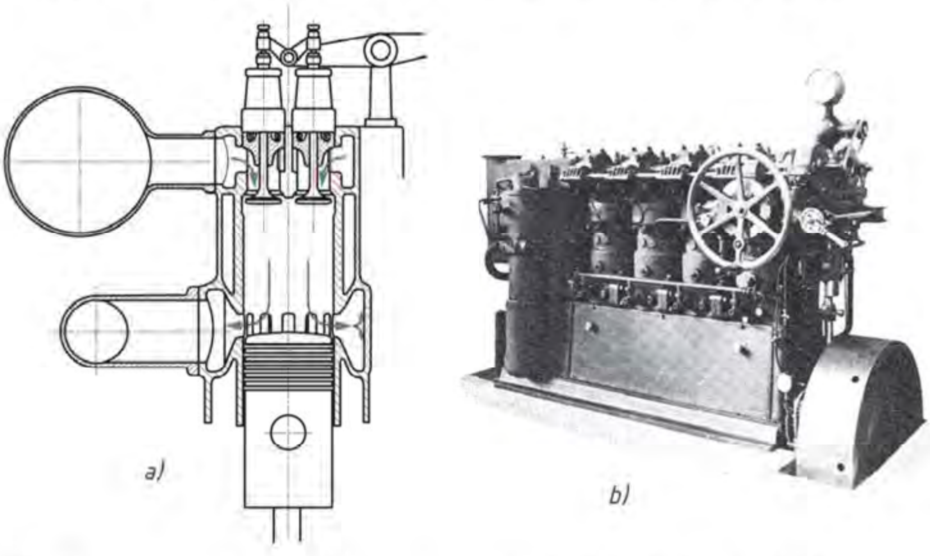

Figure 4.11 – The first two-stroke reversible marine engine built by Sulzer in 1905: *a* – general structure and gas exchange diagram; *b* – general view (adapted from [5])

Another type of spool gas distribution is a scheme in which the functions of the spool are performed by a working piston, which, when moving along the sleeve, sequentially opens or closes the ports for releasing exhaust gases and admitting a fresh charge. Sulzer developed such engines, which completely abandoned the use of valves, by 1909. The first ocean-going vessel to use such engines was the freighter «Monte Penedo», which was equipped with two four-cylinder Sulzer 4S47 crosshead bore 470 mm and stroke engines 680 mm (Fig. 4.12). The engine featured transverse loop blowing through two rows of ports cut in the lower part of the cylinder liner. At a rotation speed of 160 min^{-1}, each engine developed a power of 625 kW.

Focusing on the production of powerful marine diesel engines, the company considered the two-stroke cycle as the most rational way to increase specific power and reduce dimensions, and the ab-

4.5.4 Valve timing

Valve mechanisms is most widespread on marine four-stroke engines due to its comparative simplicity of design and reliable operation. In engines of this type, the opening and closing of the intake and exhaust ports is carried out using valves located in the cylinder cover.

On the Fig. 4.12 shows the general structure of the valve gas distribution mechanism, which assembly of valve units and a transmission mechanism.

The valve assembly includes: valve 1 (inlet or exhaust), valve seat 2, guide sleeve 3, spring 4 and its lower and upper plates 5.

The transmission mechanism in most cases consists of: levers 6, rods 7, pushers 8 and a camshaft with a set of cam washers 9.

The valves are lifted by the action of the camshaft cams, and seated and held closed by the valve springs.

In a number of four-stroke engine designs, a hydraulic transmission is used to drive the valves, in which the drive cam acts through a pusher on the hydraulic drive pump plunger called *an actuator*. The actuator creates oil pressure in the valve drive system, under the influence of which the actuator pistons move down, opening the valves (Fig. 4.16). Otherwise, this mechanism is similar to that shown in Fig. 4.15.

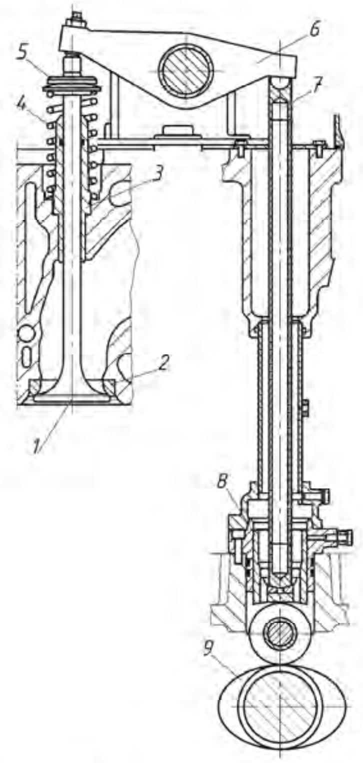

Figure 4.15 – Valve-type gas distribution mechanism of diesel engine TBD 645, Deutz MWM: 1 – valve; 2 – valve seat; 3 – guide sleeve; 4 – spring; 5 – upper valve plate; 6 – double-arm lever; 7 – pusher rod; 8 – pusher; 9 – camshaft (adapted from [7])

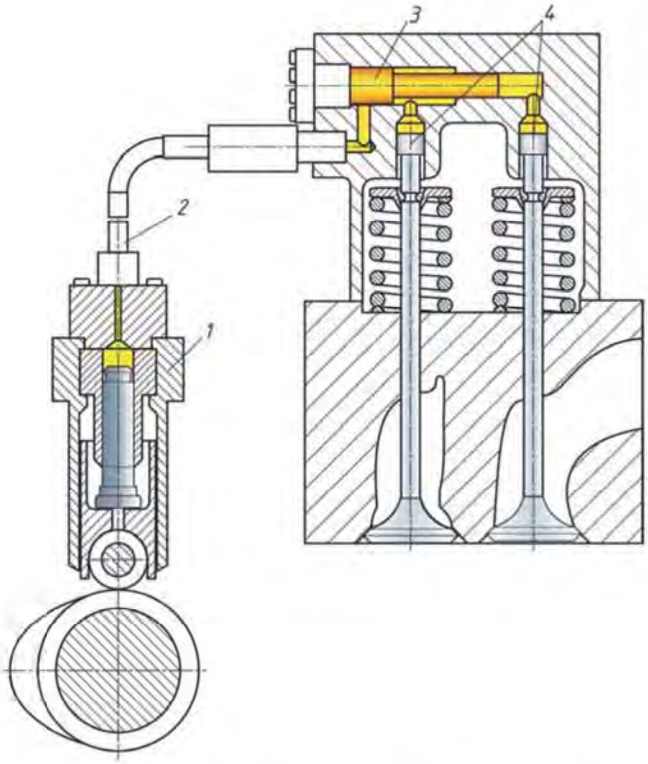

Figure 4.16 – Diagram of the gas distribution mechanism with hydraulic valve drive of the Sulzer ZA50S engine: 1 – actuator, 2 – high-pressure oil line; 3 – oil filter; 4 – valve drive pistons (adapted from [6])

The layout of the gas distribution mechanism shown in Fig. 4.12 and 4.13 is typical for most modern marine internal combustion engines. The placement of valves with this arrangement is called *top*, and the valves are called *suspended*. The upper location of the valves allows for a compact combustion chamber, which helps reduce heat losses and improve mixture formation, especially with a central location of the fuel injector.

The quality of cleaning and filling the cylinder, other things being equal, depends mainly on the size of the flow area of the gas distribution elements. The size of the flow area, in turn, depends on

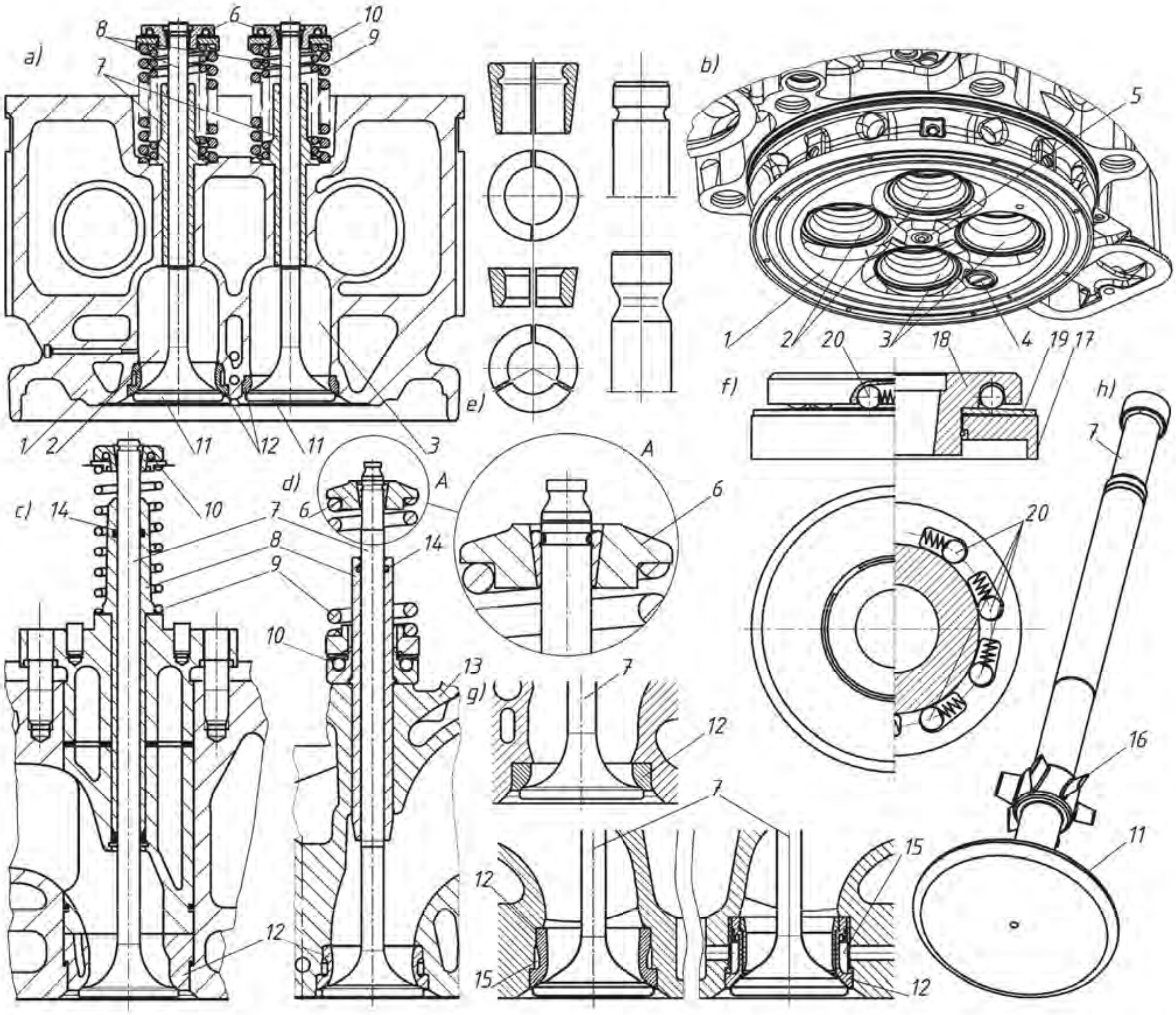

Figure 4.21 – Valve gas distribution mechanisms and their elements: *a* – cylinder cover with suspended valves of the 46 series diesel engine from Wärtsilä; *b* – fire bottom of the MaK 43C diesel engine with valve sockets; *c* – plug-in valve housing for diesel engine TBD 645, Deutz MWM; *d* – valve assembly of the MaK 25C diesel engine; *e* – conical cotters for fastening the upper valve plate; *f* – device for turning the «rotocap» valve; *g* – plug-in valve seats; *h* – exhaust valve of MAN K98MC-C diesel engine:: 1 – firing bottom; 2 – exhaust valve sockets; 3 – intake valve sockets; 4 – start valve; 5 – fuel injector spray tip; 6 – upper valve plate; 7 – valve stem; 8 – valve guide bushing; 9 – spring; 10 – rotocap; 11 – valve plate; 12 – valve seat; 13 – cover housing; 14 – sealing ring; 15 – water cooling cavity of the exhaust valve seat; 16 – impeller; 17 – rotokap stator; 18 – rotokap rotor; 19 – conical spring washer; 20 – spring-loaded balls (adapted from [7, 12, 13, 14, 15])

In the upper part of the valve stem there is a groove for installing conical wedge-shaped fasteners called *crackers* (Fig. 4.21 *e*), which fit into a cylindrical or conical groove on the housing of the rod. With their help, a valve spring thrust washer called *a plate is attached to the shank of the rod* (Fig. 4.21 *d*).

The valve spring ensures that the valve closes and presses tightly against the seat when closed. One end of the spring rests on the valve housing or cover, and the other rests against the upper valve plate. On most modern MSEs and HSEs, rotary caps are installed under the springs, mounted in the upper plate of the valve spring (Fig. 4.21 *a*) or installed between the spring and the supporting surface of the cylinder cover or valve assembly housing (Fig. 4.21 *d*).

In the mounted state, the spring must have stiffness sufficient to ensure kinematic connection between the elements of the valve drive mechanism. In marine internal combustion engines, the most

The valve drive cams must ensure their rapid opening and closing with acceptable inertia forces. This is achieved by choosing the appropriate profile, which sets the law of change in the flow area of the valve slit, the magnitude of the valve stroke, the moments of its opening and closing (Fig. 4.25).

Cam profiles are usually made symmetrical. For reversible diesel engines, this makes it possible to ensure the same conditions for cleaning the cylinder in forward and reverse modes. On non-reversible diesel engines, there are designs with asymmetrical cam profiles. In four-stroke reversible diesel engines, two sets of cam washers are installed – one for forward, the other for reverse.

In low-speed and some medium-speed diesel engines, removable cam washers are installed on the camshafts (Fig. 4.25 *a*, *d*, *e*), which have a press fit with the shaft or are a split design with coupling bolts (Fig. 4.25 *e*). During a press fit to remove the cam, as well as to adjust its position, oil is pumped into the gap between the parts under high pressure, which creates an oil cushion between the surfaces that allows the cam to move relative to the shaft. To control the position of the cam, an angular scale is applied to it, and a mark is made on the shaft (Fig. 4.24 *e*). The cam for the indicator drive usually consists of two parts connected by two fitting bolts (Fig. 4.24 *a*).

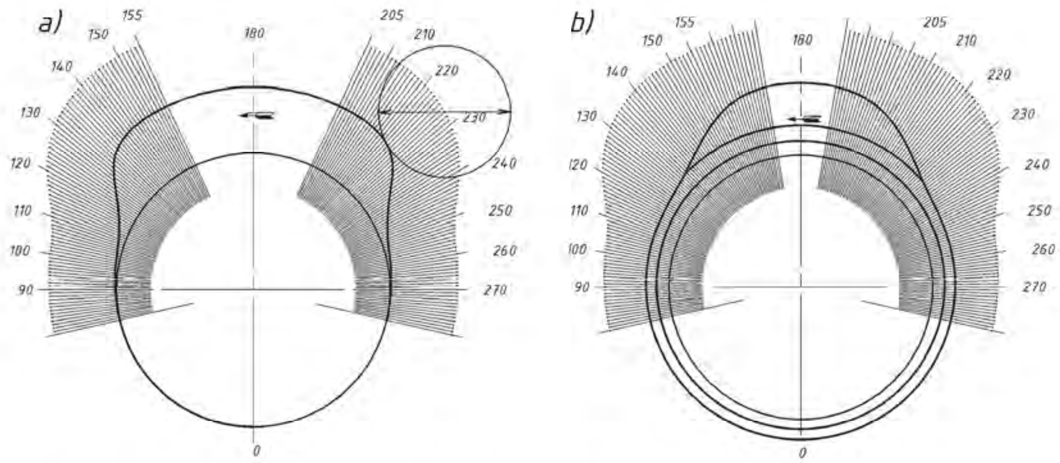

Figure 4.25 – Profiles of valve drive cams: *a* – exhaust valve of a two-stroke LSE; *b* – four-stroke MSE (adapted from [14, 19])

The camshafts of the LSE and some MSEs, together with fuel pumps, actuators and indicator drives, are assembled in the form of separate units mounted on the engine frame (Fig. 4.26 *a*). To lay the shaft in each pump block, a bed with covers is bored out, in which replaceable thin-walled support and thrust liners are installed, covered with a layer of anti-friction material. The outer liners serve to limit the axial and radial movements of the shaft, all the others – only for radial ones. A section of the shaft housing is used as a supporting surface, which is processed accordingly. As a rule, one pump unit is installed on two working cylinders.

In MSE and HSE cams and flanges are usually made as one piece with the shaft of a separate section (Fig. 4.26 *b*), connecting them directly to each other or through a cylindrical spacer insert. The developed cylindrical generatrix of connecting flanges or inserts is usually used as the supporting surface of the camshaft.

To increase rigidity, the shaft bed is usually bored into the engine block (Fig. 4.26 *b*, *c*, *d*), placing it in a special compartment for in-line and V-shaped engines or in the camber of the cylinder block for V-shaped ones (Fig. 4.26 *d*). Replaceable cylindrical bushings coated with antifriction material are installed in the bored sockets, acting as support journal bearings (Fig. 4.26 *b*). The camshafts are secured against axial movement using thrust bearings, which are usually located on the drive side. When using helical gears, this makes it possible to reduce the influence on the valve timing of thermal extensions of the camshaft and engine frame, and when using a chain drive, it avoids additional loading of sprockets and chains by axial forces.

Solid shafts are secured with caps for ease of installation. In this case, the shaft bearings are made in the form of two liners (Fig. 4.26 *c*).

The axles of all sprockets and chains are lubricated with oil using injectors.

The transmission mechanism serves to transmit force from the camshaft to the valves. In marine MSEs and HSEs, mechanical transmission mechanisms are most widely used. The kinematic diagram of the transmission mechanism largely depends on the selected engine layout and, first of all, on the location of the camshaft. Engines with an overhead camshaft are characterized by the simplest kinematic schemes, when the cam acts on the valve through a single (Fig. 4.26 *e*) or double-arm lever, as well as directly on the upper valve plate, which acts as a pusher (Fig. 4.26 *f*). The minimum number of parts makes it possible to reduce the mass of moving parts, reduce the inertial forces acting in the drive, and reduce the dimensions of the engine. For this reason, this scheme is typical mainly for water supply systems.

Most shipboard MSEs and HSEs are characterized by the transmission mechanism arrangement shown in Fig. 4.15.

The main disadvantage of a mechanical valve drive is the presence of gaps to compensate for thermal expansion of its parts, which prevent the possibility of incomplete closing of the valve. Due to the presence of gaps, the movement of the valve begins and ends with an impact, causing additional stress in the seat and disc, spring and on the contact surfaces of the transmission mechanism. The size of the gaps largely depends on the size of the drive parts and their number; the longer the parts and the more of them, the more they expand and the larger the thermal gap must be provided. In trunk engines, where the distance between the camshaft axis and the rocker arm axis is relatively small, the simplicity and reliability of the mechanical drive are decisive factors when choosing the type of transmission mechanism. To eliminate the above drawback, some HSEs and MSEs use devices that allow the clearances in the mechanical drive to be reduced to zero during engine operation without the risk of incomplete closing of the valves.

For LSEs with a large height, the mechanical drive parts can be quite large, which requires significant thermal clearances. In addition, a large mass of parts leads to an increase in inertia forces in the drive. Repeated attempts to get rid of gaps and associated shock loads in LSE drive mechanisms led to a significant complication of the design. This was one of the main reasons why all manufacturers of engines of this class switched to hydraulic transmission mechanisms.

The hydraulic valve drive is used in some MSEs (Fig. 4.15, 4.19).

The elements of the mechanical transmission mechanism are shown in Fig. 4.28. Next, their design and operating conditions will be discussed in more detail.

The pushers receive the force from the cam washers and transmit its axial component to other elements of the transmission mechanism, and lateral forces to the guide surfaces of the frame. To reduce the inertia forces of the moving parts of the transmission mechanism, the drive parts must have as little mass as possible. For these purposes in HSEs use pushers with a cylindrical guide surface in the form of hollow cups (Fig. 4.28 *a*) or T-shaped (Fig. 4.25 *b*) pushers, with a flat or spherical working surface.

To ensure more uniform wear of the guide and working surface, as well as to reduce friction forces, the pushers are given a rotational movement. To do this, the axis of the pusher with a flat working surface is shifted by 1.5...3 mm relative to the cam (Fig. 4.28 *d*), and with a spherical working surface the cam is made slightly conical (Fig. 4.28 *e*). Due to this, when the working surfaces of the cam and the pusher interact, a torque arises that rotates the pusher housing at a certain angle.

To ensure shock-free operation and reduce noise, some water pumps use pushers with a hydraulic thermal gap compensator, the general layout of which is shown in Fig. 4.28 *c*. The housing of such a pusher houses a hollow, tightly fitted piston, the upper part of which is closed by a plug with a hole for the heel of the pusher rod, and a non-return plate or ball valve is installed in the lower part. The closed internal cavity of the piston communicates through a system of grooves and holes in the housing of the housing and the piston itself with the engine oil line. Between the bottom of the pusher housing and the lower part of the piston, a closed cavity is formed in which a spring is installed, under the action of which the piston tends to occupy the upper position limited by the stop in the housing. The spring acting on the piston is several times weaker than the valve springs, so it

Gas distribution ports are divided into exhaust and purge ports. They are made in the form of rectangles, parallelograms, trapezoids, as well as oval or round. The number and location of ports depends on the chosen gas exchange scheme and the diameter of the cylinder.

Modern LSEs use a direct-flow valve purge scheme, which assumes the presence of only purge ports, which are located along the entire circumference of the cylinder in its lower part (Fig. 4.30 *a*). When choosing the shape and size of ports, they strive for their larger total width, reducing the thickness of the lintels between them, as far as strength conditions allow. This makes it possible to significantly reduce the height of the ports, thereby reducing the share of lost working stroke on the gas exchange organs. The total width of the ports for a direct-flow purge scheme is usually 0.60...0.75 of the cylinder perimeter and the height is 0.05...0.09 of the piston stroke. The largest flow area is provided by rectangular ports (Fig. 4.30 *b*), but stress is concentrated at their corners. For this reason, the upper and lower edges of the ports are made rounded, with a rounding diameter equal to the width of the port (Fig. 4.30 *c*), and in a number of designs, ports are made as several intersecting drillings (Fig. 4.30 *d*). To reduce the resistance to the incoming air flow, the inlet edges of the purge ports are slightly rounded.

In relation to the plane perpendicular to the cylinder axis, the purge ports are positioned tangentially (Fig. 3.9 *e*) at an angle of 15...25° or radially.

The tangential inclination to the cylinder axis helps to create turbulence of the fresh charge and improve the mixture formation process; however, at significant air velocities at the entrance to the cylinder, the bulk of it can be pressed by centrifugal forces against the cylinder walls. In this case, a so-called dead zone is formed along the axis, with inactive contents.

Radially directed ports have the simplest design and technology and manufacturing, however, along the jumpers, wall dead zones can also form here, which is associated with the uneven distribution of fresh charge along the perimeter of the cylinder. To eliminate this drawback, purge ports are sometimes used, the vertical axis of which is deviated from the cylinder axis by an angle of 7...10° (Fig. 4.30 *e*). Thus, the upward flow is evenly distributed over the entire surface of the cylinder liner.

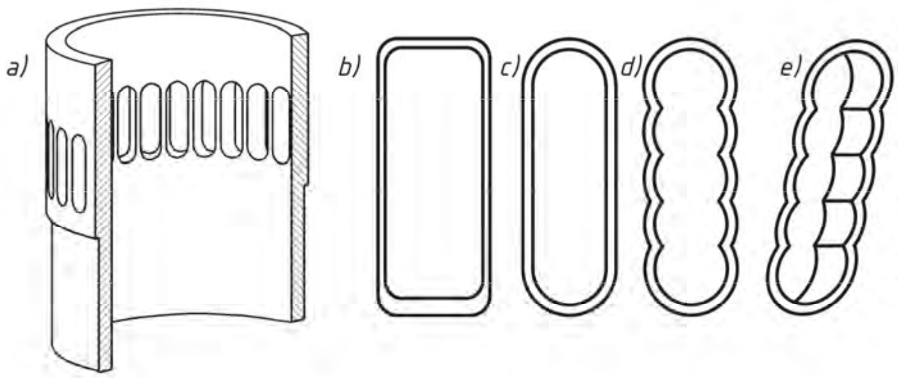

Figure 4.30 – Gas distribution ports of two-stroke diesel engines with direct-flow purge: *a* – location of ports on the cylinder liner; *b* – rectangular; *c* – with rounded ends; *d* – segmental; *e* – segmental with a tilt of the vertical axis

Two-stroke engines with direct-flow slot scavenging have, in addition to scavenging ports, exhaust ports. This scheme is used in diesel engines with oppositely moving pistons, in which air enters through purge ports, and exhaust gases exit through exhaust ports located along the entire circumference in the opposite part of the cylinder. Purge ports, as a rule, have a tangential location, and outlet ports have a radial location.

4.5.7 Hydrostatic transmission mechanisms

Hydraulic transmission mechanisms have become prevalent in LSE and some MSE. Their main advantage is the ability to transmit large forces with relatively small dimensions and masses of structural elements. This makes it possible to reduce the inertial forces acting in the drive, increase its reliability, and reduce operating noise.

The hydraulic drive consists of a piston pump-actuator driven by a cam washer, a hydraulic

of the engine; in particular, there is no need for camshafts and their drive. In addition, it makes it possible to quite easily change the operating modes of individual elements of engine systems. In particular, in relation to the gas distribution system, the hydraulic drive allows, depending on the operating mode, to change the valve timing and the law of valve movement.

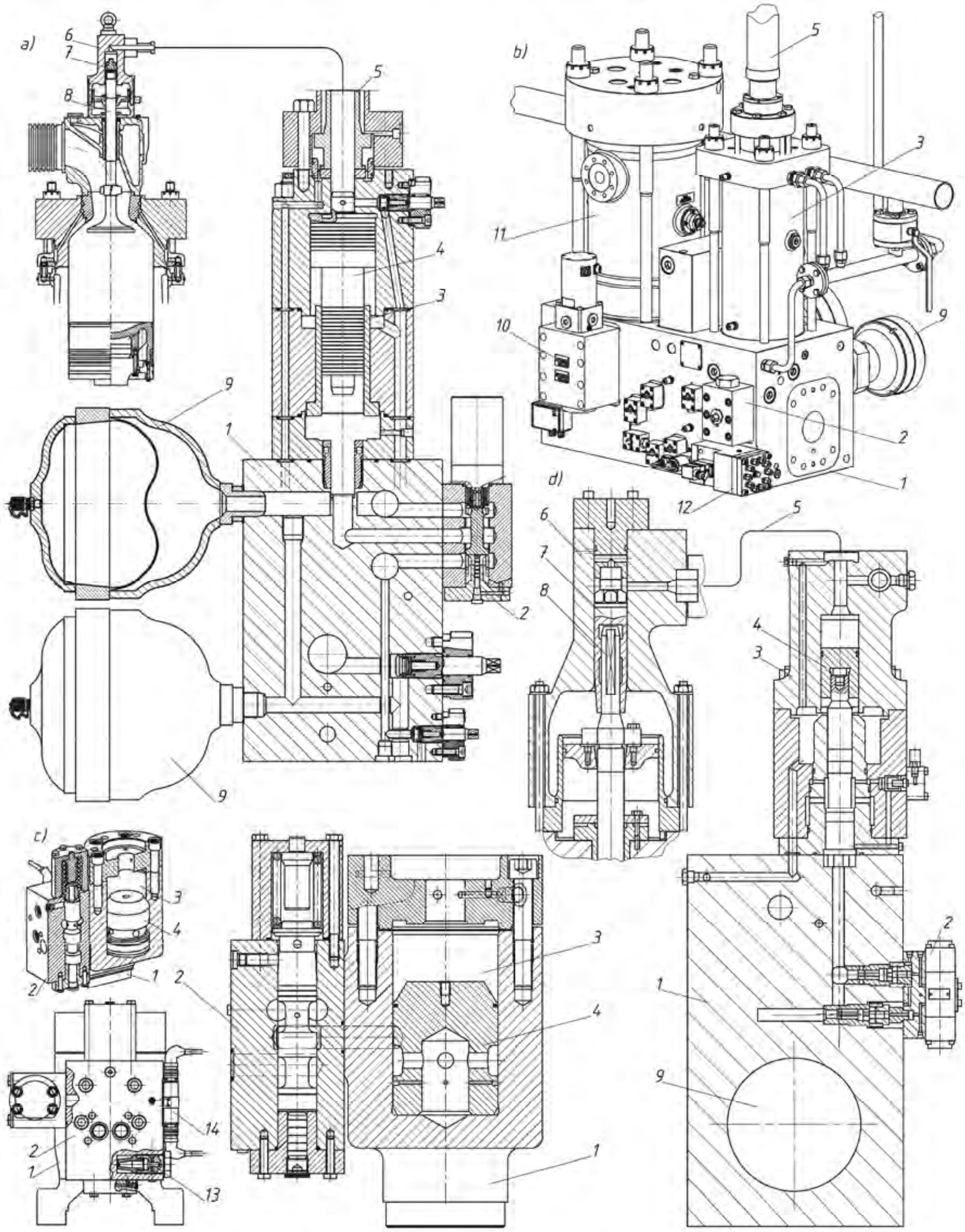

Figure 4.34 – Hydraulic drive of gas distribution mechanism actuators: *a* – engines ME series K, L, S from MAN; *b* – general view of the hydraulic block of MAN ME engines from the valve drive actuator side; *c* – general view and longitudinal section of the hydraulic actuator for diesel engines of the RT-flex series from Wärtsilä; *d* – engines of the UEC Eco-Engine series from Mitsubishi: 1 – hydraulic unit housing; 2 – control spool for the exhaust valve actuator; 3 – hydraulic cylinder; 4 – actuator piston; 5 – high pressure hydraulic line; 6 – hydraulic cylinder of the valve drive; 7 – valve drive piston; 8 – exhaust valve stem; 9 – pressure accumulator; 10 – spool control of the fuel pump actuator; 11 – fuel pump actuator; 12 – lubricator block; 13 – control oil filter; 14 – block of electrically controlled valves (adapted from [25, 26, 27, 28])

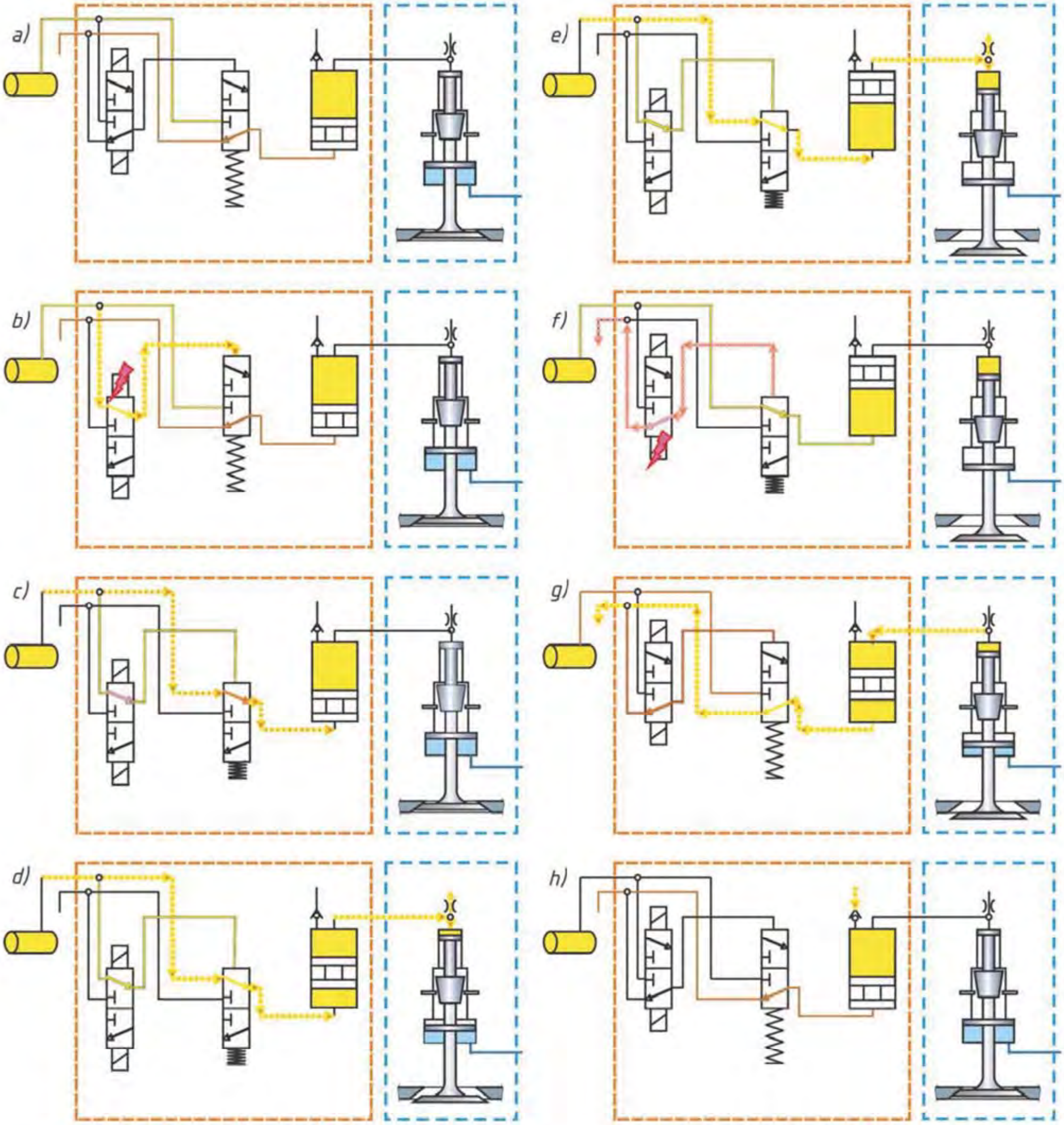

Figure 4.37 – Hydraulic diagram of the exhaust valve drive of a diesel engine of the RT-flex 96C series from Wärtsilä and the order of its operation: *a* – the valve is closed, there is no control action; *b* – control action on the opening of the main valve; *c* – the actuator spool moves and opens the oil supply to the hydraulic cylinder; *d* – the actuator piston moves, the exhaust valve opens; *e* – full opening of the valve; *f* – control action for closing the main valve; *g* – draining oil from the hydraulic cylinder; *h* – replenishment of oil leaks: 1 – control hydraulic line; 2 – main valve; 3 – actuator valve spools; 4 – hydraulic piston of the actuator; 5 – hydraulic piston of the valve drive; 6 – measuring cone; 7 – valve stem position sensors; 8 – pneumatic piston; 9 – exhaust valve; 10 – compressed air supply; 11 – deaeration throttle; 12 – drain line; 13 – make-up valve (adapted from [30])

To overcome the force from the gas pressure in the working cylinder of the engine at the first stage of opening the exhaust valve, a lot of force is required. This is achieved by the fact that the oil entering the executive hydraulic cylinder through a channel in the drive housing is supplied to the main hydraulic cavity located above the upper end surface of the spool piston (Fig. 4.36 *a*), while overcoming the spring force of the valve closing speed limiter 10. Through the holes 4 (Fig. 4.36 *e*) the oil enters the internal channel in the spool piston and then enters the additional hydraulic cavity,

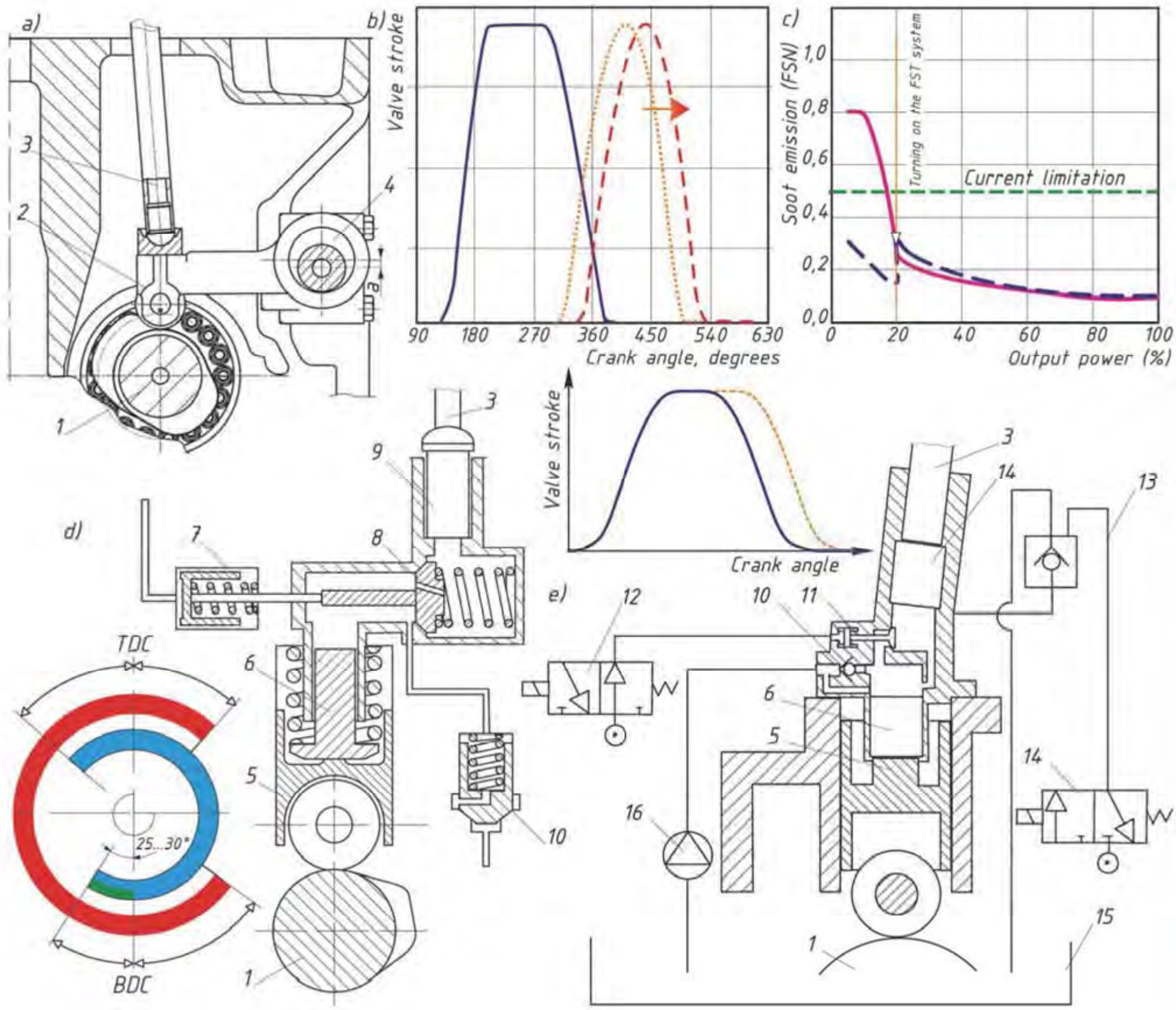

Figure 4.40 – Mechanisms for changing valve timing: *a* – with movement of the lever pusher relative to the cam washer (MaK engines); *b* – change in the angles at which the intake valve begins to open and close in MaK engines; *c* – reduction in exhaust gas smoke with increasing dynamic purging phase in MaK engines; *d* – hydromechanical drive with fluid throttling of Sulzer ZA40S engines; e – hydromechanical drive with control valve for Wärtsilä 46F series engines: 1 – camshaft; 2 – roller pusher; 3 – rod; 4 – eccentric of the lever axis; 5 – pusher; 6 – working piston; 7 – pneumatic drive of the separation valve; 8 – separating valve; 9 – executive piston; 10 – make-up valve; 11 – control valve; 12 – control valve spool; 13 – deaeration valve; 14 – drain valve; 15 – oil storage tank; 16 – charging pump (adapted from [14, 31, 32, 33])

One cylinder contains a piston that receives forces from the pusher and transmits them to the liquid. The second cylinder contains an actuator piston, which receives forces from the fluid and transfers them to the valve drive rod. At medium and high loads, the valve separating the cavities of both pistons is kept open and does not interfere with the oil circulating freely between them. When switching to low load mode, the air holding the valve is released, and the valve sits on its seat, separating the piston cavities. In this case, when the pusher moves upward, the valve opens under pressure and almost freely allows oil to pass into the cavity of the actuating piston. During the reverse stroke, the valve closes and the oil from the cavity of the slave cylinder returns to the working cylinder through the throttling channel in the valve disc. As a result, the valve closure occurs with a delay, which leads to an increase in the time it takes to fill the working cylinder of the engine with fresh charge.

Wärtsilä has developed a device for the series engines 46F, the diagram of which is shown in Fig. 4.40 *f*. Structurally, this device is similar to the previous one, with the only difference being

The share of other types of particles (dust, soot) in the total volume of air is disproportionately small. Air contamination primarily depends on the location of its intake, as well as on the volume of the engine room, its saturation with mechanisms, the frequency of air circulation, and the specific power of the mechanisms per 1 m^3 volume of the engine room.

Air purification is carried out using air purifiers, which can be divided into two groups with dynamic action and filtering action.

Dynamic purifiers include devices that use differences in the mass of these particles and air molecules to remove polluting particles. Their operation is based on the principle of the influence of dynamic forces on particles during a sharp change in the direction of the flow or when it twists. Air molecules, being lighter, manage to change the direction of their movement, and polluting particles, being heavier, continue to move by inertia and, hitting the surface of the purifier, lose speed, sticking to it. In a number of designs called *cyclones*, the air is given a rotational motion, in which a centrifugal force acts on the particles, pressing them against the wall of the purifier. Dynamic ones include baffle cleaners with an oil bath, direct-flow and counter-flow cyclones, and mesh cleaners. The advantage of such devices is their design simplicity and reusability, as well as low input resistance. In addition, such purifiers, having a relatively large volume, smooth out air pulsations, reducing suction noise and improving the operating conditions of the supercharger compressor. The disadvantage of dynamic cleaners is their relatively low cleaning efficiency; they are capable of retaining no more than 95% of polluting particles. The remaining particles entering the intake tract lead to its rapid clogging. Therefore, dynamic filter elements are currently used only for air pre-cleaning.

On modern engines, dry filter elements are widely used, the basis of which is the principle of trapping polluting particles when air passes through a microporous partition, which is used as felt, special cardboard or other non-woven materials. The pore sizes for a filter element made of cardboard are in the range of 2.5...3.5 μm, for non-woven material 12...15 μm, which allows you to retain 98.5...99.9% of polluting particles. As a result, the wear of CPG parts is reduced by 20...30%, and clogging of the flow part of the intake tract occurs much more slowly. The design of a dry cylindrical filter is shown in Fig. 4.45.

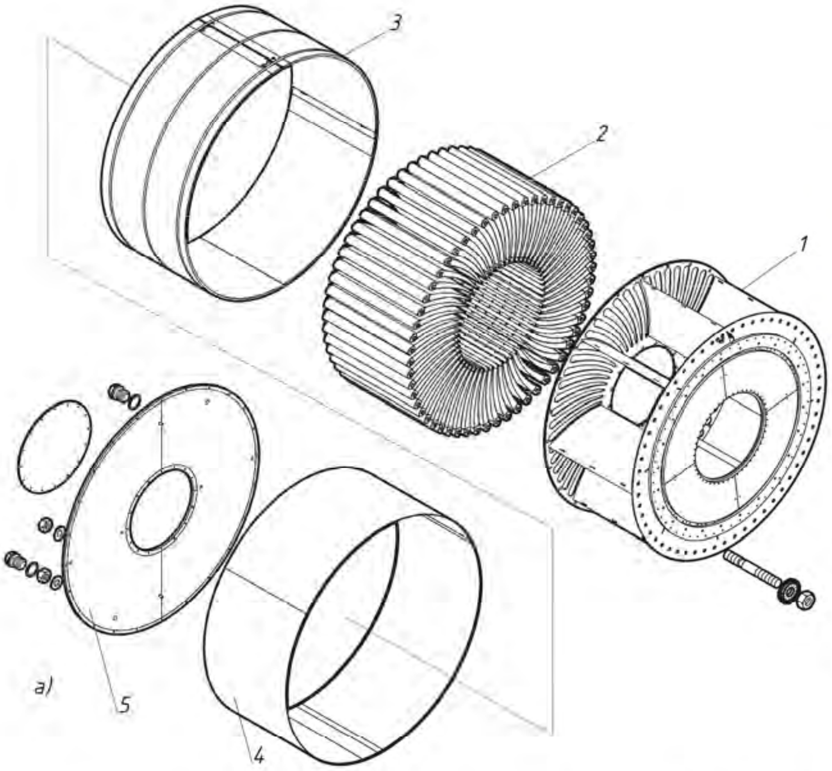

Figure 4.45 – Cylindrical dry filter element (*a*) and filter paper structure (*b*). 1 – frame; 2 – cardboard corrugation; 3 – metal mesh; 4 – felt mat; 5 – side cover (adapted from [38])

A corrugation made of special cardboard is placed on the filter frame, which allows you to increase the filter surface area many times. Along the edges the corrugation is glued to the side walls of the frame. To prevent the corrugation from collapsing, several stiffening ribs are made on the

- supercharging systems with a gas turbine compressor drive, in which the gas turbine uses the energy of the exhaust gases and drives a compressor located with it on the same shaft, forming a single unit called a *turbocharger*. In this case, the turbocharger shaft does not have a mechanical connection with the engine shaft (so-called free turbocharging) (Fig. 4.48 *c*), or is connected to the engine through a mechanical transmission (Fig. 4.48 *d*);

- combined pressurization using both drive and gas turbine compressors (Fig. 4.48 *e*).

Early marine engine developments used mechanical charging systems using both positive displacement and dynamic compressors. The presence of a mechanical connection between the compressor and the crankshaft makes it quite easy to match the characteristics of the compressor with the flow characteristics of the piston part of the engine, especially in transient operating conditions. In a number of designs, not only a rigid but also a flexible mechanical connection was used for this purpose, allowing the compressor speed to be changed, regardless of the crankshaft speed. The main disadvantage inherent in this method of supercharging is a decrease in the mechanical efficiency of a diesel engine associated with the expenditure of 7...10% of the energy generated by the engine to drive the compressor, and as a result, increased fuel consumption. In addition, a significant increase in air pressure required unjustified complication of the design of both the drive and the compressor itself.

Due to these shortcomings, all leading manufacturers of four-stroke marine diesel engines have switched to using free turbocharging as the main way to increase engine power and efficiency.

In two-stroke crosshead engines, combined charging schemes were used for a long time, when sub-piston cavities were used as one of the stages, operating as piston pumps (Fig. 4.44 *f*). Subsequently, with the improvement of gas turbine charging systems, the use of sub-piston cavities was abandoned.

Currently, two-stroke LSEs use a combined charging scheme, which includes a turbocharger that supplies the engine with air in medium and full power modes, as well as electrically driven blowers that provide operation in start-up modes and low loads (Fig. 4.44 *e*). In addition, blowers ensure diesel operation in emergency modes, for example, in the event of a turbocharger failure.

Thus, gas turbine supercharging today is an integral part of modern marine diesel engines. Many years of design and operation experience have shown that increasing the mass of the air charge is the most efficient use of exhaust gas energy. Other methods of supercharging are currently used extremely rarely or in combination with the same gas turbine supercharging.

4.7.3 Gas turbine charging

Gas turbine supercharging or turbocharging is a method of supercharging based on the use of energy from the exhaust gases of the piston part of the engine. The main unit is a turbocharger that has no mechanical connection with the engine shaft. The use of gas turbine charging makes it possible to increase engine power by 1.5...4 times.

The widespread introduction of gas turbine supercharging in marine diesel engines is explained by a number of advantages that these engines have compared to naturally aspirated ones:

- at a constant rotation speed, the use of boost makes it possible to increase engine power almost in proportion to the increase in boost pressure;
- the use of supercharging can significantly improve the weight and dimensions of both the engine itself and the ship's power plant as a whole;
- increasing the air pressure at the engine inlet using turbocharging leads to an increase in the share of useful work in the overall energy balance, resulting in an increase in engine efficiency and a decrease in specific fuel consumption (Fig. 4.49);
- the presence of a gas turbine significantly reduces the exhaust noise;
- inflatable engines are less sensitive to changes in external conditions, primarily to changes in temperature, pressure and humidity of the surrounding air;
- with the same method of organizing the operating process, supercharged engines have better environmental performance.

The disadvantages of turbocharging are the increase in mechanical stress of the engine and the

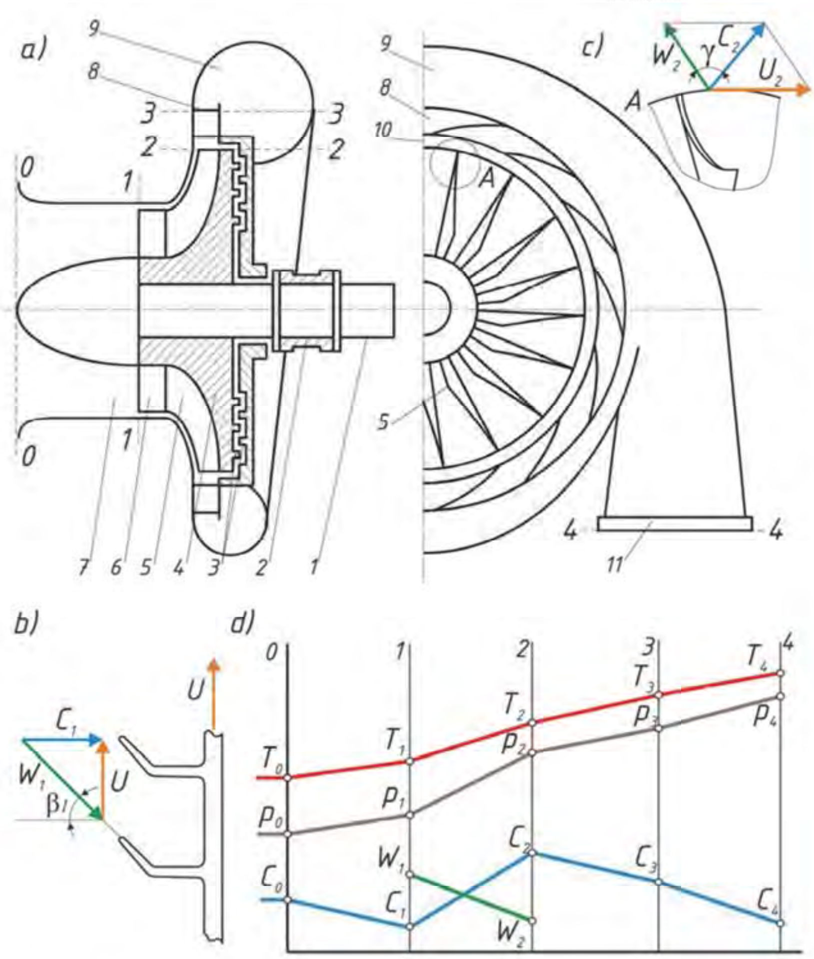

Figure 4.53 – Diagram of a radial-axial centrifugal compressor (*a*), speed distribution at the inlet to the impeller (*b*) and at its outlet (*c*), as well as changes in gas parameters in the main sections of the compressor (*d*): 1 – drive shaft; 2 – bearing; 3 – labyrinth seal; 4 – impeller hub; 5 – impeller blades; 6 – rotating guide vane; 7 – input device; 8 – diffuser blades; 9 – snail-type air collector; 10 – bladeless section of the diffuser; 11 – outlet pipe

The air, having passed through the filter, enters the inlet device, which serves to increase the stability of the flow and is tapered. Thus, the inlet device ensures a uniform supply of air to the wheel with minimal losses. The compressor impeller is mounted on the same shaft as the gas turbine impeller. It is a disk with end radial blades forming expanding interscapular channels. The diffuser is made in the form of an expanding channel located between the walls of the housing. It has a bladeless section, which is an annular gap between the end of the impeller and the diffuser blades, and a blade section formed by a system of blades located in a circle around the compressor wheel, with expanding channels in the direction of air movement. On small turbochargers, the blade section of the diffuser may be missing. The outlet pipes are expanding channels located in a circle around the diffuser. They collect the air leaving the diffuser and direct it into the engine intake pipes. Often the outlet pipes are made in the form of a so-called «snail».

To explain the principle of operation of a centrifugal compressor, let us consider the movement of air in its elements, identifying the most characteristic sections (Fig. 4.53 *a*): *0-0* – cross section at the inlet to the compressor. Air parameters in this section: pressure p_0, temperature T_0 and speed c_0; *1-1* – cross section at the entrance to the impeller (p_1, T_1, c_1); *2-2* – exit from the impeller (p_2, T_2, c_2); *3-3* – outlet from the diffuser with air parameters p_3, T_3, c_3; *4-4* – output from the compressor with air flow parameters p_4, T_4, c_4.

When a centrifugal compressor operates, a vacuum is formed at the entrance to the impeller, under the influence of which air (cross section *0-0*) with parameters p_0, T_0, c_0 enters the input device.

Air enters the impeller with an absolute speed c_1, pressure p_1 and temperature T_1. Rotating, the compressor wheel involves the air entering it into rotational motion (section 1-1), with a peripheral (transferable) speed U_1 (Fig. 4.53 *b*). Next, the flow moves in the inter-blade channels of the wheel at a speed W_1, called the relative speed, equal to the geometric difference between the absolute speed c_1 and the circumferential speed U_1. In order to ensure a smooth (impact-free) entry of air into the impeller, it is necessary that the relative velocity vector W_1 coincides with the direction of the leading edges of the wheel blades (angle β_1). The leading edges of the wheel blades, bent to a cer-

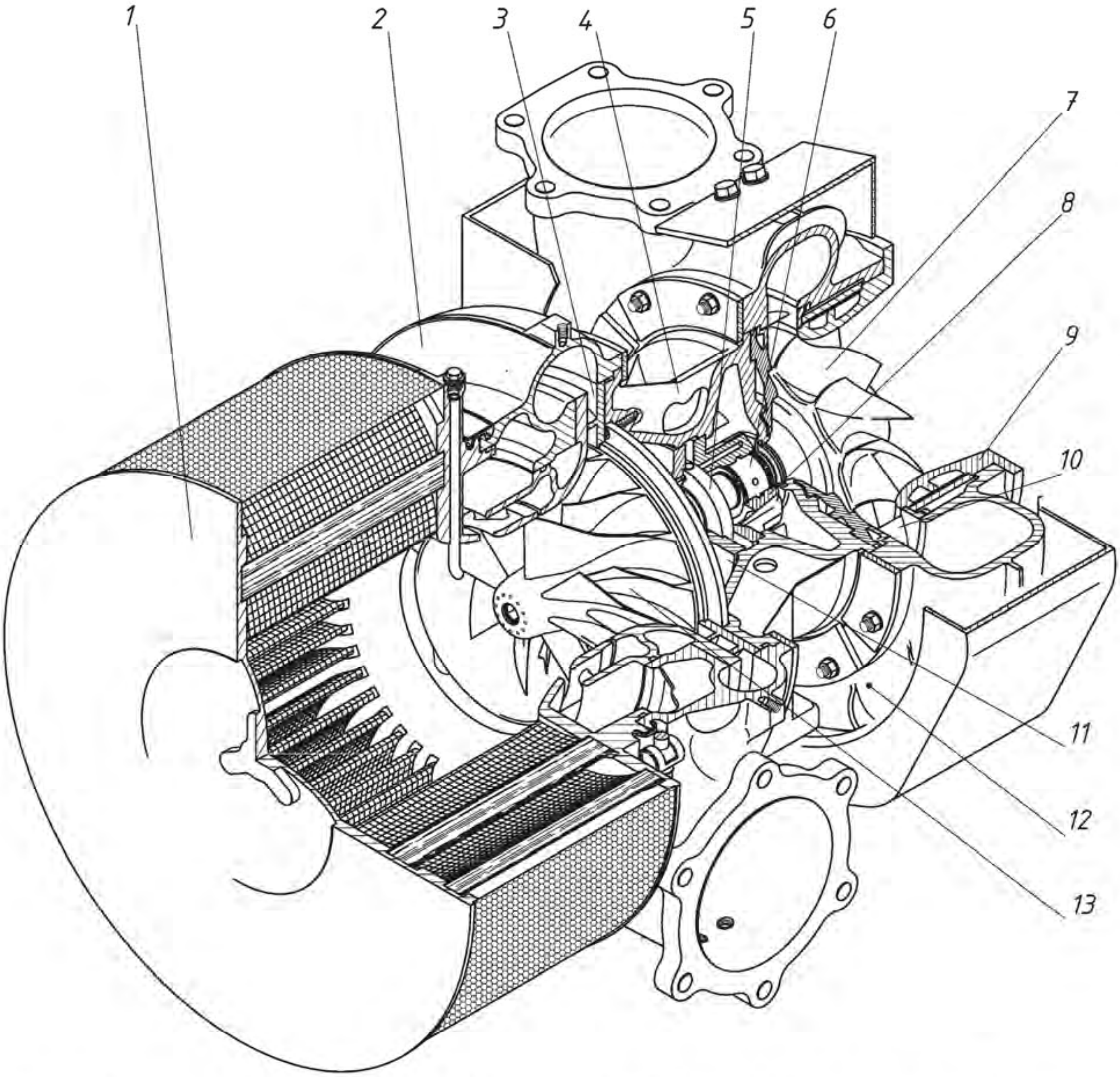

Figure 4.57 – Turbocharger with radial turbine type TPS52-F from ABB: 1 – air filter; 2 – compressor outlet; 3 – diffuser; 4 – bearing housing; 5 – axial bearing; 6 – radial bearing on the side of the turbine wheel; 7 – turbine wheel; 8 – bearing unit; 9 – turbine deflector; 10 – turbine guide vane; 11 – seal on the compressor side; 12 – turbine outlet pipe; 13 – compressor wheel (adapted from [45])

4.7.4.4 Design features of individual elements of turbocharger units. All turbochargers consist of fixed elements and a movable rotor, which in turn consists of turbine and compressor wheels, as well as the shaft connecting them. As a rule, the selected rotor design determines the design of the entire turbocharger unit as a whole, including housing elements. In modern marine internal combustion engines find use of rotor designs in which the turbine and compressor wheels are located at different ends of the shaft connecting them. In this case, the bearings on which the shaft rests can be located both along its edges and between the impellers.

The most commonly used rotor designs are shown in Fig. 4.58.

For turbochargers made according to the scheme shown in Fig. 4.58 *a*, bearings and seal assemblies are located at the edges of the rotor (the so-called cantileverless design). With this arrangement, the bearings on the turbine side are the furthest from the disk and are not exposed to high temperatures.

Bearings are usually mounted in separate housings at the edges of the unit, so they are easily accessible for lubrication, maintenance and replacement. The bearings used have a small diameter, so their peripheral speed are relatively low, which, in turn, reduces heat generation and reduces the

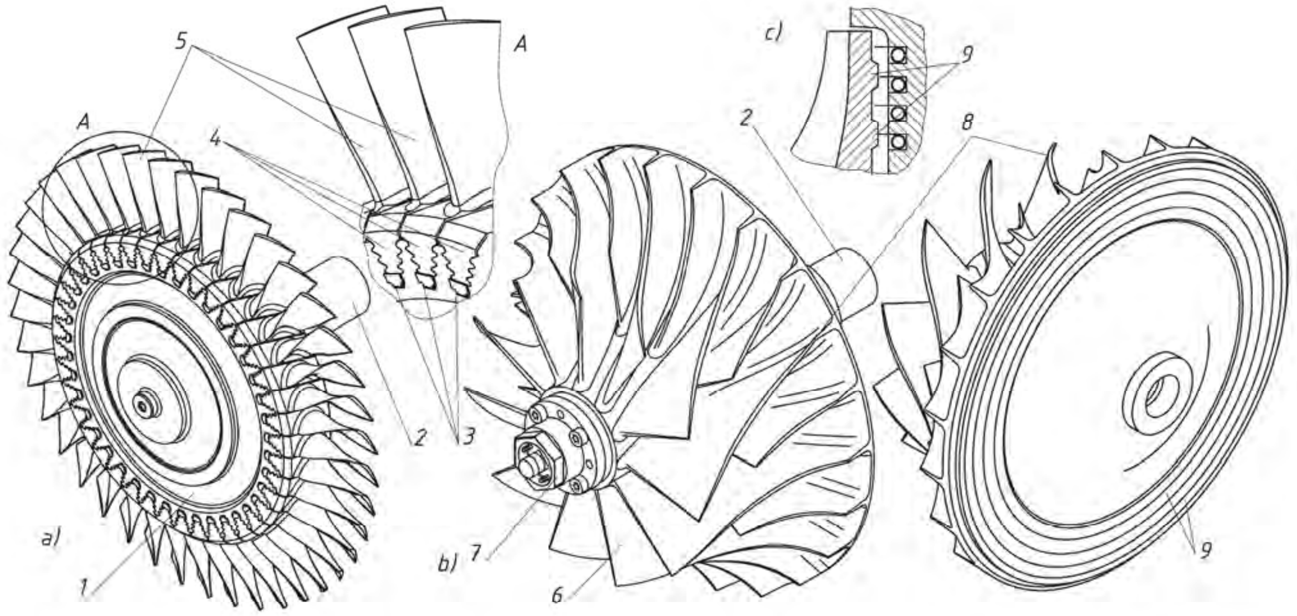

Figure 4.61 – Turbine wheel (*a*) and compressor wheel (*b*) of a TCA type turbocharger from MAN. Labyrinth seal of the compressor wheel (*c*): 1 – turbine wheel hub; 2 – rotor shaft; 3 – locking strips for fastening the blades; 4 – fastening of the blade shank of the «christmas tree» type; 5 – turbine blades; 6 – compressor blades; 7 – compressor wheel fastening nut; 8 – compressor wheel hub; 9 – protrusions of the labyrinth seal (adapted from [48])

The turbine inlet guide vane plays an important role in shaping the characteristics of both the turbine and the entire turbocharger unit as a whole. In the inlet guide vane, an initial increase in the speed of the gas flow occurs and the angle of its entry onto the turbine wheel blades is formed. As has already been shown above, deviation from this angle significantly increases energy losses from the impact of the gas flow on the turbine blades, as a result of which the efficiency of the turbine drops sharply.

The turbines of modern MICE supercharging units use guide vane devices with both fixed and movable blades.

Devices with fixed guide vane arrays are simpler in design however they provide high turbine efficiency only in a very narrow range of operating conditions. Structurally, a fixed inlet guide vane usually consists of outer and inner rings, between which blades are, installed (Fig. 4.62).

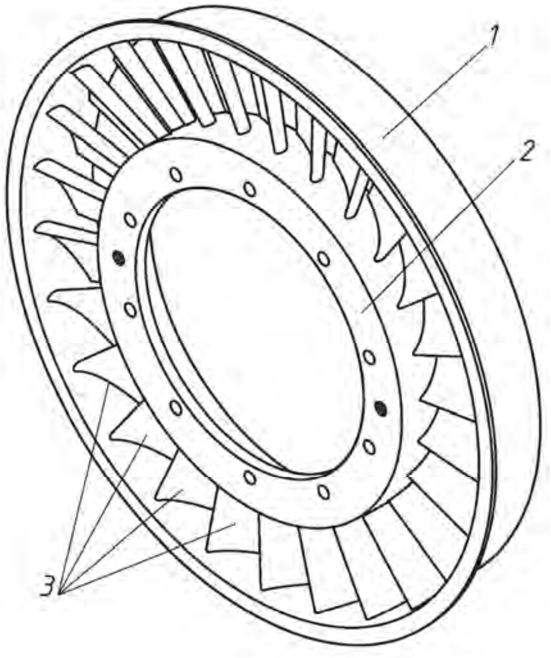

Figure 4.62 – Inlet guide vane of an axial turbine of a TCA type turbocharger from MAN with a fixed blade grid: 1 – outer ring; 2 – inner ring; 3 – blades (adapted from [48])

oil to the thrust surface, oil supply channels are milled on the inside of the bearing. Oil supply bevels can also be made on the thrust heel (Fig. 4.64 *f*). To compensate for distortions and equalize loads on the thrust surface, the latter can be installed elastically through a package of thin gaskets, the gaps between which are filled with oil.

4.7.4.7 Design features of stator elements.

Turbocharger stator consists of three main units – the compressor housing, the gas outlet housing, the turbine housing, which are connected to each other by flanges and centered by landing collars (Fig. 4.65).

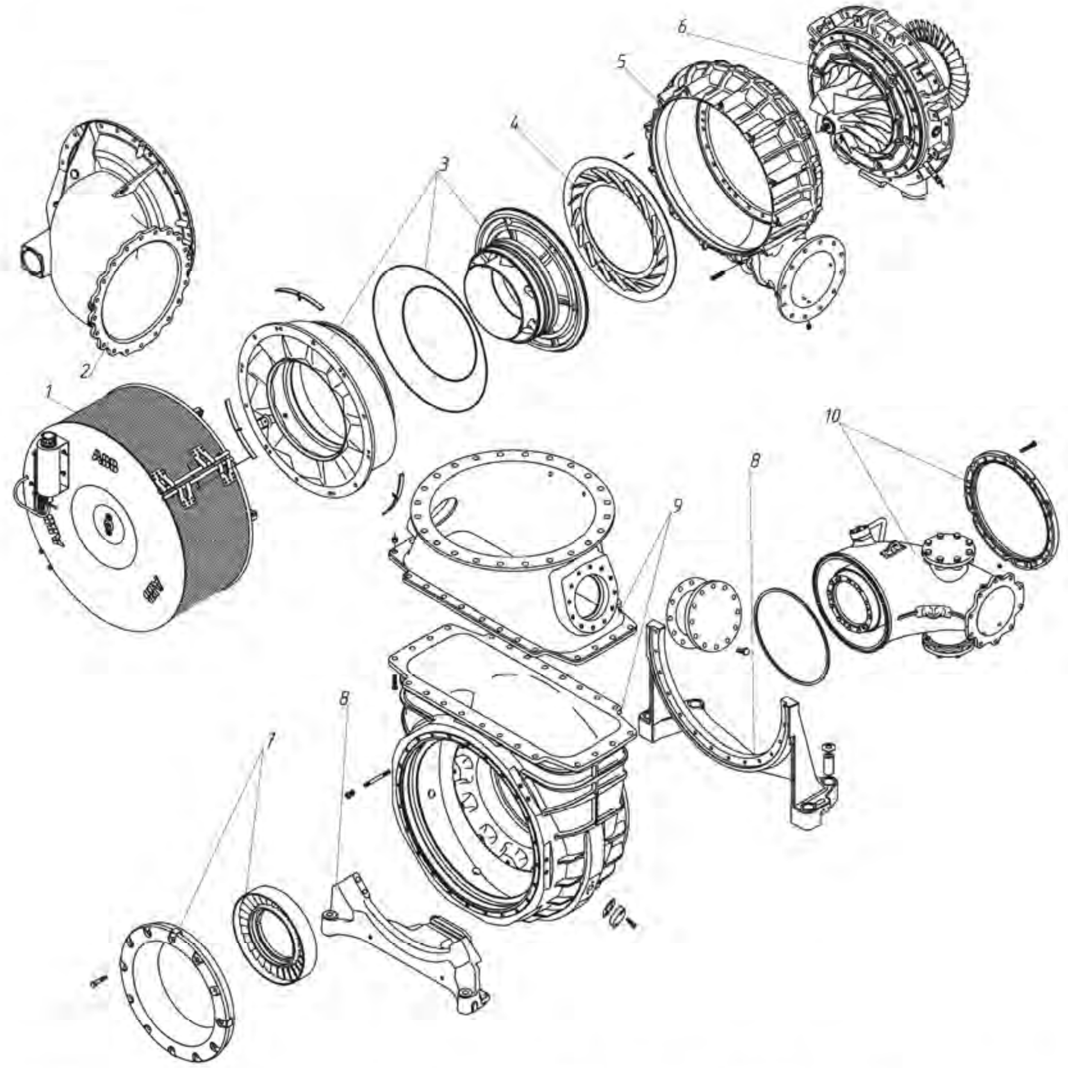

Figure 4.65 – Housing parts of a turbocharger type TPL76-C from ABB: 1 – air supply pipe; 2 – air filter; 3 – details of the sealing disc; 4 – compressor diffuser; 5 – air exhaust spiral chamber (volute); 6 – middle housing with bearings and rotor; 7 – details of the turbine inlet guide vane; 8 – turbocharger mounting supports; 9 – gas turbine housing with gas outlet pipe; 10 – details of the gas supply channel (adapted from [44])

The assembly of housings can be carried out in different relative positions of individual elements, which expands the layout possibilities of using the units. The support brackets located on the turbine housing can also be rotated relative to the turbine housing, which further facilitates the installation of the turbocharger on the engine.

Inlet and outlet pipes are connected to the housing parts.

The design of the housing parts has a decisive influence on the performance of the turbocharger. The air entry into the compressor should be smooth, without sharp turns. The supply pipe and the muffler filter must not distort the air velocity fields at the inlet.

fuser at gas flow rates that differ significantly from the calculated values. It should be noted that flow stall from the compressor blades occurs both with a decrease and with an increase in gas flow compared to its calculated value; however, the stall mechanisms and their impact on the operation of the compressor are significantly different. In Fig. 4.69 shows three cases of flow around the leading edges of the impeller blades. If in the case shown in Fig. 4.69 *c*, the resulting vortex zone is blocked by the oncoming flow in the near-wall region of the blades, then in the case shown in Fig. 4.69 *a*, the vortex zone has the opportunity to develop along the flow part of the compressor wheel, since the walls of the blades tend to expand the vortex zone. As a result, vortex zones with reduced pressure can form along the entire flow path of the compressor wheel, through which a counterflow of air periodically occurs from the air receiver, where it is under increased pressure, in the direction of the inlet pipe.

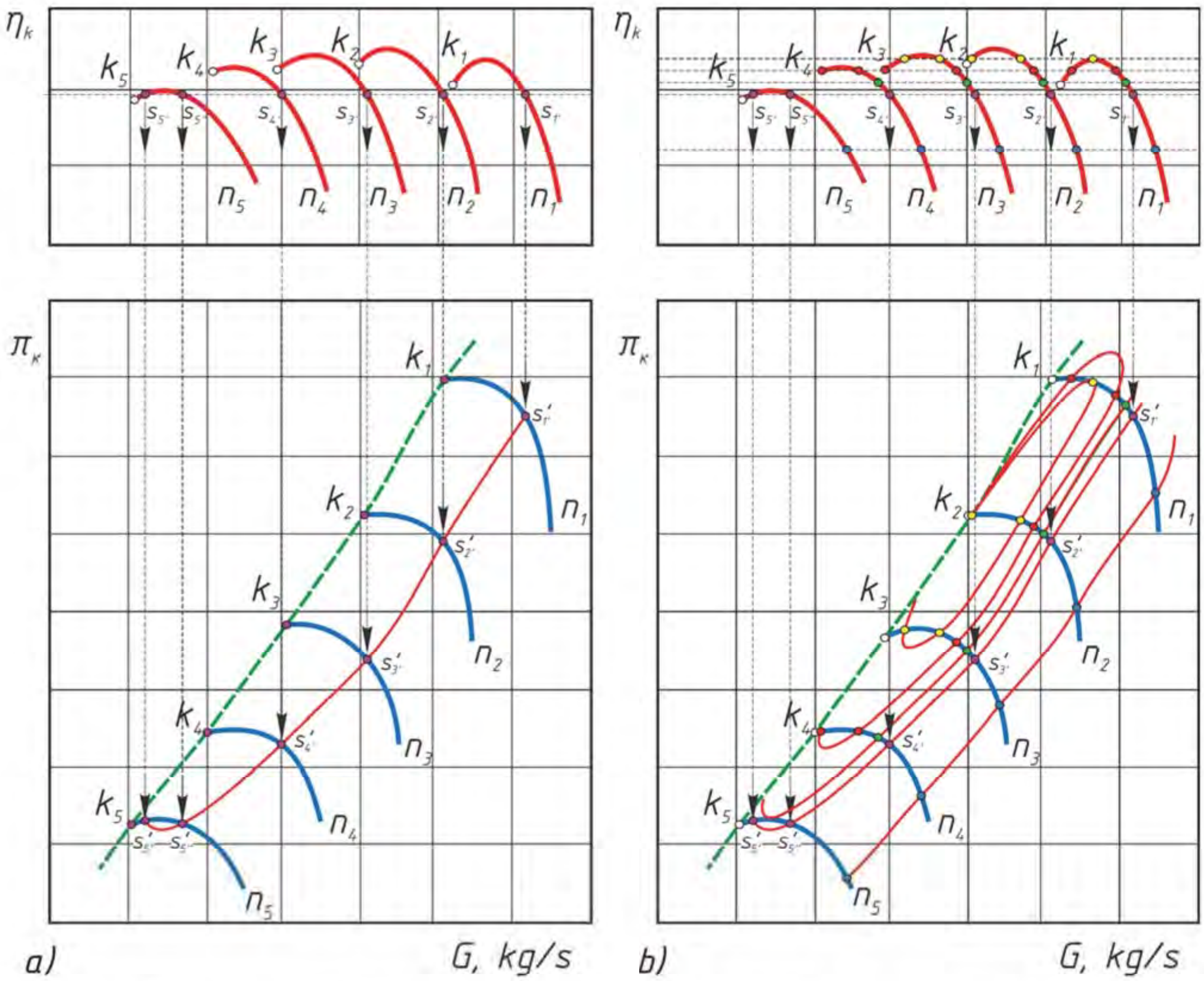

Figure 4.70 – Construction of a combined compressor characteristic by transferring adiabatic efficiency isolines to the flow-pressure part of the characteristic

When installing a vaned diffuser in a compressor, surge may occur on its blades. The mechanism of occurrence is shown in Fig. 4.71.

There are three possible cases of flow around the diffuser blades. When the speed of gas exit from the wheel corresponds to the calculated one (Fig. 4.71 *a*), the air flow expands evenly along the entire inter-blade space. With an increase in flow rate and, therefore, speed c_2, vortex zones begin to form along the convex side of the blades (Fig. 4.71 *b*), which are blocked by the oncoming flow. At low gas flow rates (reduction in speed c_2), vortex zones form at the concave walls of the blades (Fig. 4.71 *c*). The gas flow, moving along a flat trajectory, is rejected from the concave

regulation can be more effective than bypassing, but the design of turbochargers is significantly more complicated. The geometry can be changed either stepwise or smoothly by rotating the guide vane array blades.

An example of a radial turbine with a stepwise change in the settings of the turbine guide vane is shown in Fig. 4.74.

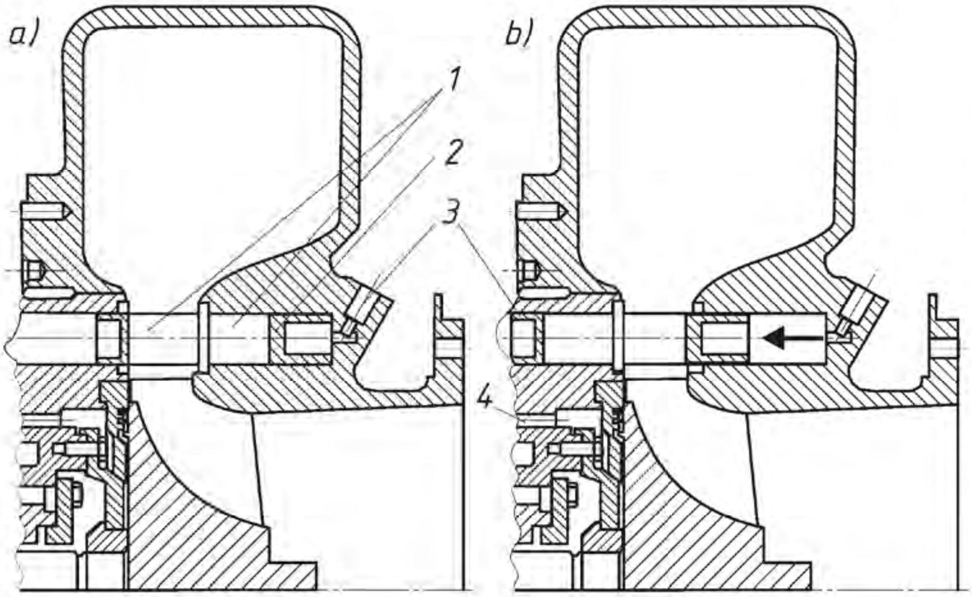

Figure 4.74 – Turbocharger with a stepwise change in the geometry of the guide vane apparatus: *a* – partial load mode; *b* – mode close to full power. 1 – sets of blades with different flow sections; 2 – pneumatic piston; 3 – control air supply channel; 4 – sealing air supply channel

When working at low loads, a guide vane apparatus with a small flow area is used, and when moving to full loads, a device with large flow sections is used. Both sets of blades are located in the turbine housing and are moved by a pneumatic piston depending on the engine load.

Mitsubishi Heavy Industries has developed its own version of stepwise regulation of turbochargers when switching to partial load modes for its MET series turbochargers. This technology is called Variable Turbine Inlet (VTI). The basic operating principle of the system is shown in Fig. 4.75 and 4.76. For stepwise regulation, the turbine of the charging unit is equipped with a two-level guide vane grid.

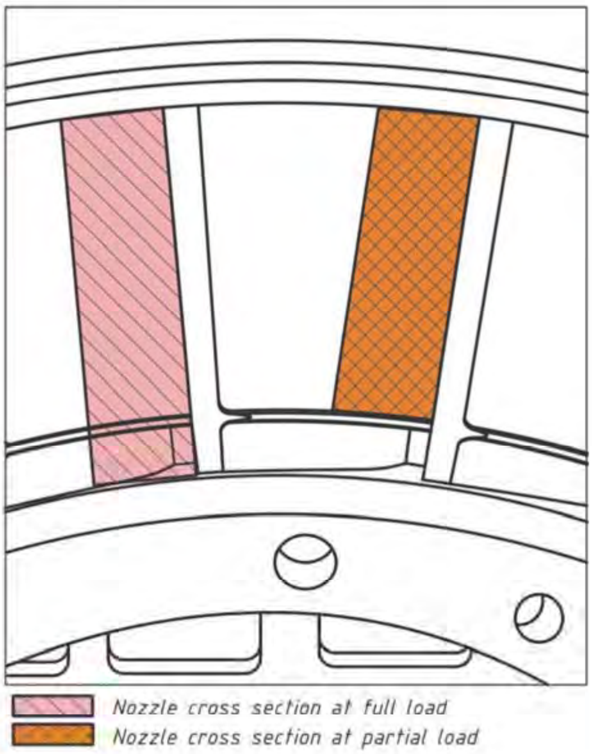

Figure 4.75 – Two-level nozzle array of MET series turbochargers from Mitsubishi Heavy Industries equipped with VTI system (adapted from [51])

An important sector for the application of controlled charging technology is gas and gas-diesel engines, in which the conditions of knock-free combustion greatly depend on the ratio of the components of the gas-air mixture.

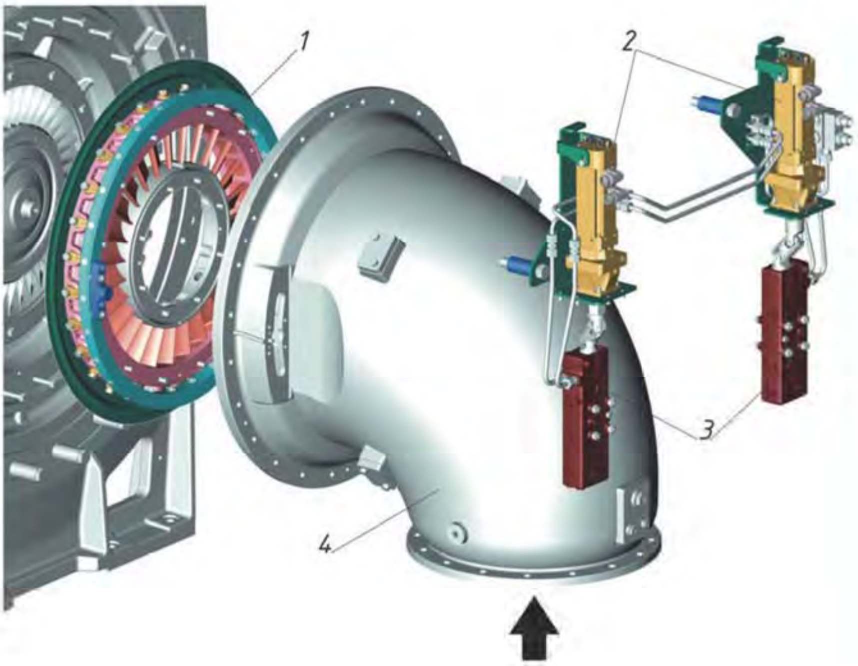

Figure 4.80 – Drive mechanism of the inlet guide vane of the axial turbine of the TCA type turbocharger unit from MAN equipped with the VTA system (adapted from [53])

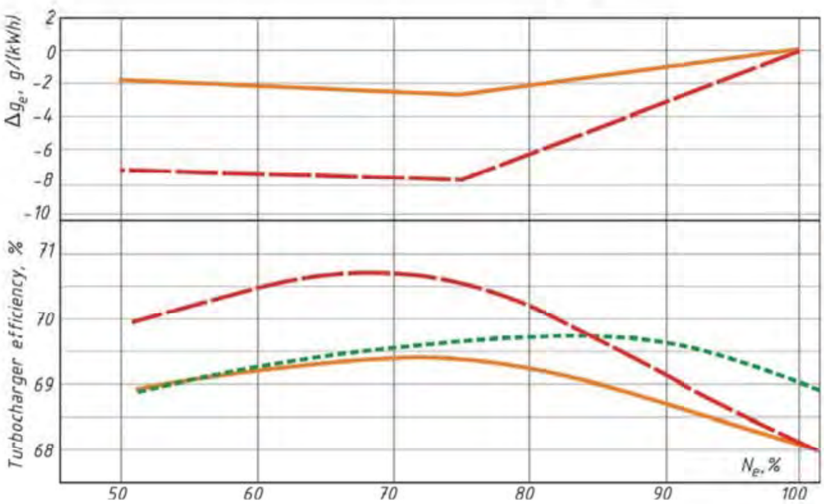

Figure 4.81 – Change in specific fuel consumption and efficiency of the TCA55-2 turbocharger unit when using VTA technology on a low-speed MAN 4T50ME-X engine. —— – characteristic with the maximum opening angle of the inlet guide vane; – – – – with the maximum closing angle of the inlet guide vane; - - - - – characteristic with a fixed inlet guide vane (without VTA) (adapted from [53])

The most difficult process is matching the characteristics of the turbocharger and the piston part of the engine during transient conditions.

The lack of a mechanical connection between the diesel crankshaft and the turbocharger rotor leads to the fact that when the load changes, the turbocharger reacts with a delay due to the inertia of both gas flows and mechanical parts. So, as the load increases, the fuel pumps increase the fuel supply to the engine, and the turbocharger increases the air supply with a lag. In this case, the optimal air-fuel ratio is disrupted, and the efficiency of fuel combustion decreases. Engine operation is accompanied by smoke.

In some designs, to reduce the acceleration time of the inflating unit, compressed air is supplied to the compressor blades from the ship's general compressed air system, using additionally installed nozzles in the sealing disk (Fig. 4.82). The air supply control system is combined with the fuel pump control system. As the load increases, the speed controller tends to move the fuel rack in the direction of increasing fuel supply. In this case, the control system limits the movement of the rack and at the same time opens the solenoid valve supplying air to the compressor blades. As the turbocharger spins up, the control system removes restrictions on the position of the fuel rack and reduces the air supply until it completely stops.

Electric blowers are used in combined supercharging systems of low-speed engines in start-up and light load modes.

Radial-axial type fans are used as air blowers (Fig. 4.86).

The blowers turn on automatically when the engine starts or when the boost pressure drops below the set norm. To automatically separate the cavities of the air receiver and the blowers, movable, hinged air dampers are used.

4.10 Engine intake tract elements

Elements of the engine intake tract include air pipes, charge air temperature stabilizers, moisture separators and intake receivers.

Air pipes are metal boxes of round, rectangular or transitional cross-section (for example, with a transition from round to rectangular) through which air is supplied to various elements of the intake tract. Connecting flanges are installed at the ends of the gas ducts (Fig. 4.87).

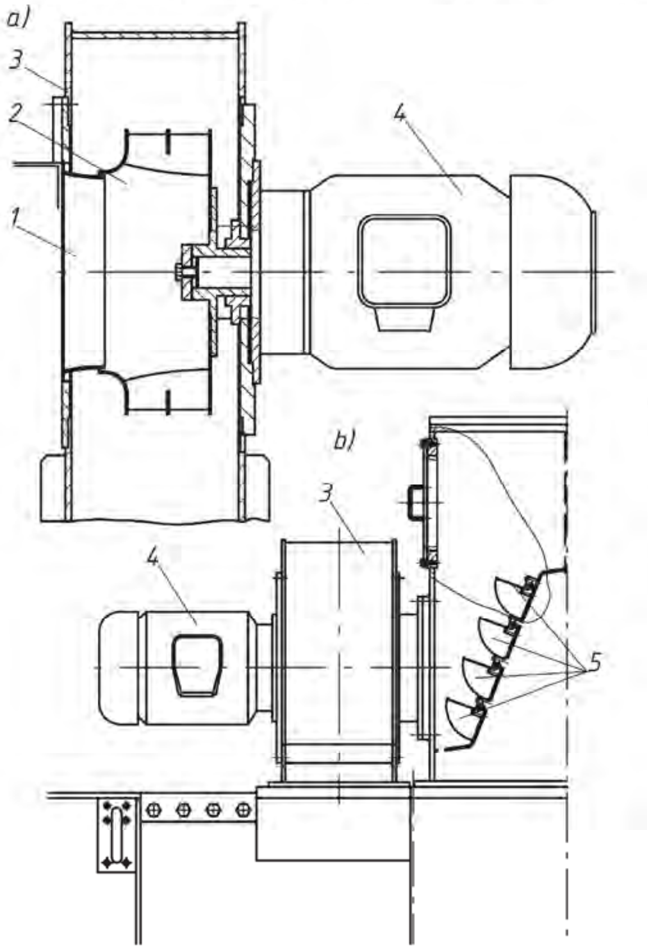

Figure 4.86 – Electric blower (*a*) and its inclusion in the air supply system of a diesel engine type RTA 72 from Wärtsilä (*b*): 1 – inlet deflector; 2 – rotor; 3 – housing; 4 – electric motor; 5 – check valves (adapted from [11])

In some cases, air ducts are equipped with thermal expansion compensators made in the form of metal bellows. If temperatures allow, expansion joints can be made in the form of spacers made of elastic non-metallic composite materials.

Charge air temperature stabilizers. As noted earlier, a side effect of the process of compressing air in a compressor is a significant increase in its temperature. In some cases, its value can reach 160...210°C. If air is supplied to the engine at such temperatures, the weight filling of the cylinders will be less, since the air density decreases with increasing temperature. In addition, the thermal stress of the cylinder-piston group parts will increase and the environmental performance of the engine will deteriorate. In this regard, before air is supplied to the cylinders, it is pre-cooled. A decrease in the charge air temperature for every ten degrees increases the mass of the charge entering the working cylinder by 2...2.5% and leads to a decrease in the average operating cycle temperature at increased boost pressure.

For example, on low-speed and medium-speed engines, a receiver design made in the form of a cylindrical cylinder closed at the ends with profiled covers has become widespread.

The receiver is connected to the charge air cooler housings by welding, and to connect it to the cylinder jackets, adapter pipes with flanges are welded.

4.11 System elements for exhaust and noise reduction of exhaust gases

The exhaust gas removal, exhaust and noise reduction system consists of an engine exhaust tract, including gas exhaust pipes and exhaust receivers, gas exhaust channels, noise mufflers, exhaust gas toxicity reduction systems and heat recovery systems. The design of the exhaust tract largely depends on the selected engine supercharging scheme, the number of turbocharger units used, and the method of their connection.

As already indicated, on marine engines, charging schemes with pulsed supply of gases to the turbine and with supply of gases to the turbine at constant pressure are used.

To implement each of these charging methods, structurally different gas exhaust tract systems are used in the area from the exhaust channels of the cylinder covers to the entrance to the gas turbine.

In Fig. 4.92 *a* is a diagram of the supercharging of a four-stroke engine with constant pressure in front of the turbine and with the supply of gases to the turbine in the form of pressure pulses (Fig. 4.92 *b*). In the first case, gases from all cylinders enter one common receiver, the volume of which is 10...15 times greater than the volume of the working cylinder. The pressure pulses from the individual cylinders are smoothed out and the gases enter the turbine at a relatively constant pressure.

With pulse charging, the cylinders are grouped in such a way that the pressure pulse in one cylinder does not overlap with the pulse in the other. To do this, individual groups of cylinders are led into separate gas exhaust pipes of a relatively small volume. As a result, gases enter the turbine in the form of pressure pulses through two or more independent channels (Fig. 4.93 *a*).

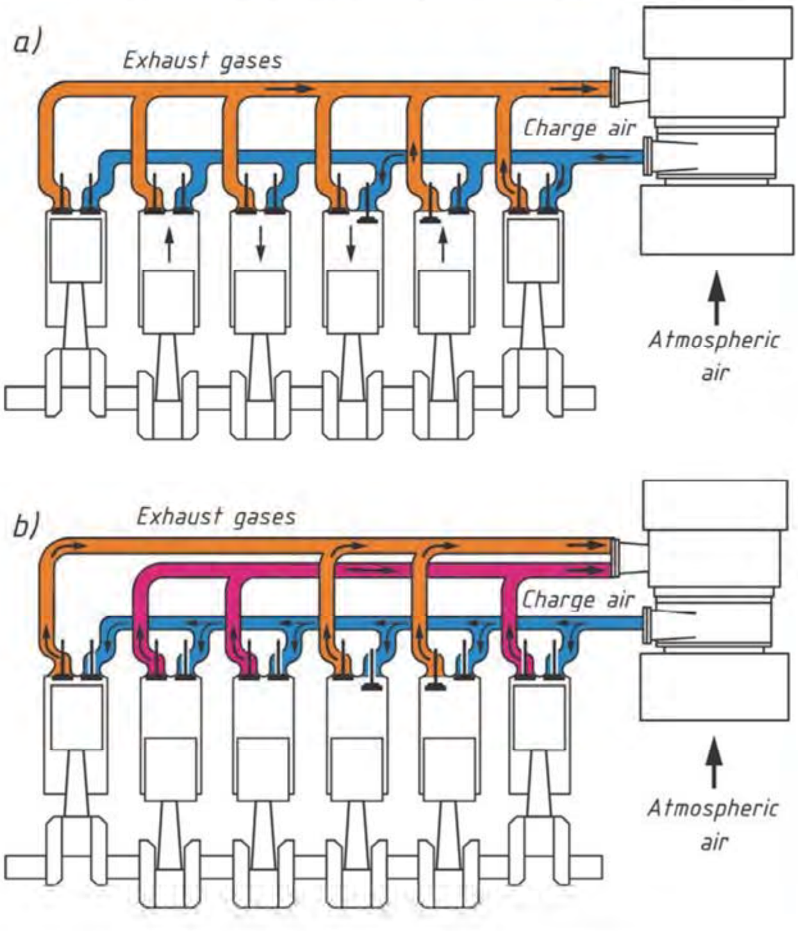

Figure 4.92 – Diagram of the organization of the exhaust tract of a four-stroke engine with constant pressure in front of the turbine (*a*) and with the supply of gases to the turbine in the form of pressure pulses (*b*) (adapted from [61])

4.13 System elements for exhaust heat recovery

The exhaust gas heat recovery system serves to transfer thermal energy to other media for the purpose of its further use or to directly convert heat into mechanical work.

The use of exhaust gas heat to heat other media is usually used for heating purposes. To do this, the gases enter heat exchangers *called recovery boilers*, in which their heat is transferred to water circulating through heated pipes. Typically, such boilers produce water steam, which is supplied for heating fuel, cargo, living and service spaces, as well as for other technological needs of the ship.

The increase in the power of modern engines has led to the fact that, despite a significant reduction in the share of exhaust gas energy in the overall thermal balance, the amount of this energy significantly exceeds the ship's needs for it. In this case, recycling systems are used to convert excess thermal energy into mechanical work.

There are three main schemes for such transformation used in modern shipbuilding:

- part of the exhaust gases is passed past the turbocharger to a recovery power turbine, which can be connected to the engine shaft through a mechanical or electrical transmission. The latest technology is called PTG (Power Turbine Generator) to implement it, an additional electric motor is installed connected to the diesel crankshaft;

- the main part of the exhaust gases after the turbocharger enters the recovery boiler, and some of them enter the boiler past the turbocharger. This makes it possible to increase the temperature of the gases at the inlet to the boiler by 50...60°C and increase its steam production. The steam produced in the boiler is supplied to a steam turbine, which, working on a generator, releases energy into the general ship network or is electrically connected to the engine crankshaft (STG technology (Steam Turbine Generator);

- part of the gases is discharged to the power gas turbine, and after it is supplied to the recovery boiler. This is where the gases come from the turbocharger. The steam produced in the boiler is supplied to a steam turbine, which is mechanically connected to the gas turbine. The energy generated by both turbines is transferred to the engine crankshaft or used to generate electricity (Combined Turbines).

The use of the first technology (PTG) makes it possible to increase its power by 4% when the engine is running at nominal mode. STG technology makes it possible to increase the power of the entire power plant by 5...7% and the combined technology makes it possible to increase the power increase to 10%. At the same time, fuel consumption for modern diesel plants with LSE is 155 g/(kWh).

The power turbines used in such installations are single-stage gas turbines of the same design as the turbines of high-efficiency turbochargers. To remove power from the turbine, a reduction gear is installed on the side of the dismantled compressor wheel.

A more detailed consideration of exhaust gas heat recovery systems is beyond the scope of this book; on this issue, one should refer to specialized literature.

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SECTION 5

Fuel systems of marine diesel engines. Mixing and combustion in diesel engines, indicators operation of fuel equipment

5.1 Marine fuels, their composition and physical and chemical properties

Liquid fuels traditionally used for marine diesel engines are products of oil refining and are conventionally made into two main types – light (Marine Diesel Oil – MDO) and heavy (Heavy Fuel Oil – HFO). The basis of all petroleum fuels are predominantly hydrocarbons of the paraffin, aromatic and naphthenic groups. The content of components of a particular group, expressed in fractions of a unit, is called *group composition of the fuel*. Each group of hydrocarbons differs from each other in the number of carbon and hydrogen atoms, the structure of the carbon skeleton and the type of bonds between the atoms, which determine the differences in their properties [1].

Hydrocarbons of the paraffin group are divided into saturated (alkanes) and unsaturated (olefins).

Alkanes (paraffins) are hydrocarbons with the general formula C_nH_{2n+2}, in the molecules of which the carbon atoms are connected to each other by single valence bonds, and the rest of their valences are extremely saturated with hydrogen atoms. Hence the name of alkanes – saturated hydrocarbons. The simplest representative of the homologous series of paraffins is methane – CH_4.

Olefins (alkenes) belong to unsaturated hydrocarbons with the general formula C_nH_{2n}. Their molecule contains one double bond. The first representative of the homologous series is ethylene C_2H_4.

Hydrocarbons of the paraffin group, especially alkanes, have the highest ability to spontaneously ignite.

Aromatic hydrocarbons are hydrocarbons whose molecules contain one or more benzene rings. The general formula is C_nH_{2n-6}. The most important representative of aromatic hydrocarbons is benzene C_6H_6, in the molecule of which six carbon atoms, connected by double valence bonds, form a regular hexagon. This structure of the molecule makes it extremely stable, making these hydrocarbons resistant to spontaneous combustion.

Naphthenic hydrocarbons groups are divided into cycloalkanes and cycloalkenes. The molecules of the former contain cycles of different sizes, the carbon atoms in which are connected to each other only by single valence bonds. General formula: C_nH_{2n}. Cycloalkenes contain one double bond and have the general formula C_nH_{2n-2}.

Naphthenic hydrocarbons, in terms of their ability to self-ignite, occupy an intermediate position between paraffins and aromatic hydrocarbons. The high content of naphthenic hydrocarbons in the fuel composition leads to an increase in its viscosity.

The approximate ratio of various groups of hydrocarbons in the composition of marine fuels is given in table. 5.1.

Table 5.1 – Group chemical composition of marine fuels

Type of fuel	Paraffins	Naphthenes	Aromatic
Marine Diesel Oil	0.30…0.58	0.05…0.15	0.30…0.50
Heavy Fuel Oil	0.05…0.50	0.40…0.70	0.10…0.25

In addition to the group composition, the elemental composition of fuels is distinguished. The approximate ratio of the main chemical elements in the composition of marine fuels is given in Table. 5.2.

Table 5.2 – Elementary chemical composition of marine fuels

Type of fuel	Carbon (C)	Hydrogen (H)	Oxygen (O)	Sulfur (S)
Marine Diesel Oil	0.86…0.88	0.124…0.128	0.002…0.006	–
Heavy Fuel Oil	0.85…0.87	0.121…0125	0.004…0.006	0.004…0.04

It is obvious that the group and elemental chemical compositions significantly affect the ability of the fuel to self-ignite. The more saturated and less aromatic hydrocarbons a fuel contains, the higher its ability to self-ignite, which is assessed by the cetane number (CN).

Cetane number (*Cetane Index*) characterizes the ability of a fuel to self-ignite. To determine it, a mixture consisting of two standard, chemically pure hydrocarbons is used: highly flammable cetane $C_{16}H_{34}$, the ability of which to spontaneously ignite is taken as 100, and difficult to ignite α-methylnaphthalene $C_{10}H_7CH_3$ – an aromatic hydrocarbon, the spontaneous ignition of which is taken as zero. By mixing these components in various ratios, you can obtain a standard mixture with any cetane number from 0 to 100 conventional units. By comparing the self-ignition of the fuel under study with a similar indicator of the reference mixture, its cetane number is determined. In other words, the cetane number of a fuel is an indicator of flammability, numerically equal to the percentage (by volume) of cetane in the reference mixture, which has the same ability to self-ignite as the fuel being tested.

Distillate fuels with a high cetane number (CN = 45...60) are used mainly for high-speed diesel engines, which have significantly less time for the combustion process. For heavy fuels used in low-speed diesel engines, CN = 25...40. Along with engine tests, calculation methods are used to assess the self-ignition of fuels based on an assessment of their group composition. For example, the CCAI (Calculated Carbon Aromaticity Index) method from Shell. The CCAI index is an ignition delay indicator, calculated based on the density and viscosity of the fuel oil.

Next, we will consider other important properties and characteristics of fuels that determine the possibility and scope of their use in marine diesel engines.

Self-ignition temperature (*Ignition Temperature; Autoignition Temperature*) – the minimum temperature at which the fuel ignites spontaneously without an external ignition source. The auto-ignition temperature depends on the group composition of the fuel and the physical characteristics of the charge in the diesel combustion chamber. For marine fuels, at an air pressure at the end of compression of 5.5...16.0 MPa, this temperature lies within 200...250°C.

Flash *point temperature* – the minimum temperature at which a mixture of vapors and air ignites upon contact with an open flame. For marine engines, this temperature should not be less than 60°C when tested in a closed crucible. This indicator characterizes the fuel from the point of view of fire safety; therefore, when storing fuel on a ship, the temperature in the tanks must be at least 10°C below the flash point. Fuels with a flash point below 60°C can only be stored on board in closed metal drums (for example, special DMX gas oil with a low cloud point).

Pour *point temperature* is the lowest temperature at which the fuel will continue to flow when cooled at specified standard temperature conditions. The pour point of distillate fuels lies in the range (-10…-20°C). For heavy fuels it can be quite high (up to +10°C). The pour point determines the possibility of pumping fuel through the pipelines of the ship's fuel system at low temperatures.

Heat of combustion (*Specific Energy*) or calorific value is the amount of heat that is released as a result of 1 kg fuel combustion. In the calculations, the lowest calorific value (Q_l) is accepted, which is less than the total calorific value by the amount of the heat of phase transition of water vapor formed during fuel combustion. For marine fuels of petroleum origin, Q_l lies in the range of 39800...44000 kJ/kg. By default, when calculating the operating process of internal combustion engines, Q_l = 42700 kJ/kg is taken.

Kinematic viscosity is *a* measure of the fluidity of a fuel at a certain temperature. It is defined as the resistance force of two layers of liquid with an area of 1 cm^2, located at a distance 1 cm from each other and moving relative to each other at a speed of 1 cm/s, per unit area. The unit of kinematic viscosity is Stokes, denoted as St (1St). In practice, in the technical specifications for fuel supply, kinematic viscosity is indicated in hundredths of Stokes – centistokes (cSt) (note: 1 cSt = 1 mm^2/s). The viscosity of the fuel decreases as it heats up, so it is indicated at a specific

7.0% by volume. The properties of DFA are close to the DMA brand, but it contains FAME.

DMZ brand fuel is identical in characteristics to DMA brand, but has a higher minimum viscosity.

DFZ brand fuel of the brand is identical in its characteristics to the DMZ brand, contains the addition of biodiesel fuel – FAME.

DMB brand fuel is similar in characteristics to DMA grade, but may contain a small amount of residual components. Commercial name: Marine Diesel Oil (MDO).

DFB brand fuel is identical in characteristics to DMB brand, contains the addition of biodiesel fuel – FAME.

Heavy marine fuels RM (residual marine fuels) are the widest group of marine fuels, which are mixtures of heavy residual fractions of oil distillation with distillate fractions (ISO 8217 standard) (Table 5.4).

Table 5.4 – Grades of marine residual fuels and requirements for them

Characteristic	Dimension	Limits	Fuel grade ISO-F-										
			RMA	**RMB**	**RMD**	**RME**	**RMG**				**RMK**		
			10	**thirty**	**80**	**180**	**180**	**380**	**500**	**700**	**380**	**500**	**700**
Kinematic viscosity at 50°C	mm^2/s	max.	10	thirty	80	180	180	380	500	700	380	500	700
Density15°C	kg/m^3	max.	920	960	975	991	991				1010		
CCAI	-	max.	850	960	860	860	870				870		
Sulfur	wt.	max.	Regulatory requirements [1]										
Flash point	°C	min.	60.0	60.0	60.0	60.0	60.0				60.0		
Hydrogen sulfide	mg/kg	max.	2.00	2.00	2.00	2.00	2.00				2.00		
Acid number	mg	max.	25	2.5	2.5	2.5	2.5				25		
Total besieged	volume, %	max.	0.10	0.10	0.10	0.10	0.10				0.10		
Carbon residue – micromethod	volume, %	max.	2.50	10.00	14.00	15.00	18.00				20.0		
Pour point (upper) [2] in winter	°C	max.	0	0	thirty	thirty	thirty				thirty		
Pour point (upper) [2] in summer	°C	max.	6	6	thirty	thirty	thirty				thirty		
Water	volume, %	max.	0.30	0.50	050	0.50	050				0.50		
Ash	volume, %	max.	0.04	0.07	0.07	0.07	0.10				0.15		
Vanadium	mg/kg	max.	50	150	150	150	350				450		
Sodium	mg/kg	max.	50	100	100	50	100				100		
Aluminium + silicon	mg/kg	max.	25	40	40	50	60				60		
Used lubricating oils (ULO): calcium and zinc; or calcium and phosphorus.	mg/kg	-	calcium > 30 and zinc > 15 or calcium > 30 and phosphorus > 15										

[1] The maximum sulfur content is determined in accordance with legal restrictions.
[2] The pour point must be suitable for the intended area of operation of the vessel.

The widespread use of heavy fuels in marine diesel engines is dictated by the fact that their cost is significantly lower than the cost of distillate fuels. However, heavy fuels have increased viscosity, sulfur content, water and mechanical impurities. For their efficient combustion in marine diesel

In an effort to reduce carbon dioxide emissions to zero, the use of ammonia (NH_3) and hydrogen (H_2) as fuel for marine engines is being considered.

It is estimated that shipping emits about 1.05 million tons of carbon dioxide per year and is responsible for almost 3% of greenhouse gas emissions worldwide. Ammonia, produced using renewable energy sources, is set to be one of the main fuels for the future of shipping.

Hydrogen can provide virtually zero greenhouse gas emissions if produced by electrolysis of water using electricity generated from renewable sources. Leading manufacturers are actively working on the use of these two promising types of fuel.

5.3 History of the creation and development of marine diesel fuel systems

The problem of developing efficient systems for supplying fuel to the combustion chamber and atomizing it arose almost simultaneously with the development of Rudolf Diesel's rational engine. Since the engine was originally intended to run on coal dust, the fuel system was designed with this fuel in mind. To supply coal dust to the cylinders, Diesel designed an air system in which fuel was blown into the combustion chamber with a stream of pre-compressed air. The first experiments revealed a number of difficulties associated, first of all, with the dosing and supply of coal dust into the working cylinders, so in further experiments the coal was replaced with lamp kerosene. Successful experiments with this fuel for a long time determined approaches to the design of fuel systems, primarily for marine diesel engines. For a long time, ship engines used the so-called compressor fuel supply system, in which compressed air was used to atomize it [6].

The advantage of this system was the relative simplicity of the design and the absence of high requirements for the manufacturing accuracy of the main elements. The disadvantage is the presence of a bulky air compressor, which consumed 10...15% of the power of the engine itself.

It should be noted that, having abandoned the use of coal dust as fuel, R. Diesel attempted to design a liquid fuel supply system with mechanical atomization, but the low level of technology at that time did not allow satisfactory results to be obtained. For this reason, in his further work the inventor was forced to return to the compressor spray system, which had proven itself well during previous experiments.

Subsequently, many designers made attempts to improve the compressor system in order to reduce its dimensions and improve the quality of fuel atomization. Attempts have been made to use water vapor obtained by recycling the heat of exhaust gases to atomize fuel, and other design solutions have also been used, but no significant progress has been made in this direction.

The first who managed to build a compressorless engine with fuel ignition by compression was the Gustav Trinkler. And although in the engine he developed, fuel was supplied to the combustion chamber using an air fuel injector, the engine did without a compressor. The air to power the injector was taken from the cylinder during the compression stroke and pressed to a higher pressure by a special piston device, which was driven by the camshaft.

Of course, the disadvantages inherent in the compressor fuel supply system pushed many engineers and inventors to search for alternative solutions, especially since acceptable designs were known and tested even before the advent of the diesel engine. Back in 1891, the English company Richards Hornsby & Sons produced plunger pumps with operating principles similar to modern ones. However, to seal the plunger in such pumps, a set of seals was used, which did not allow obtaining high enough pressures necessary for high-quality fuel atomization. Only in 1912, the American Otto Persson proposed abandoning the gland seal system, replacing them with an accurate (precision) fit of the plunger and bushing.

On diesel engines, plunger pumps developed by James McKechnie were first used in 1910 by Vickers. The pumps had a spring drive in which the energy required to move the plunger was stored using a cam mechanism by compressing the spring. At the moment corresponding to the start of the supply, the cam released the spring, which actuated the plunger.

An important aspect of choosing the type of fuel system is the ability to accurately dose a cyclic portion of fuel depending on the engine operating mode.

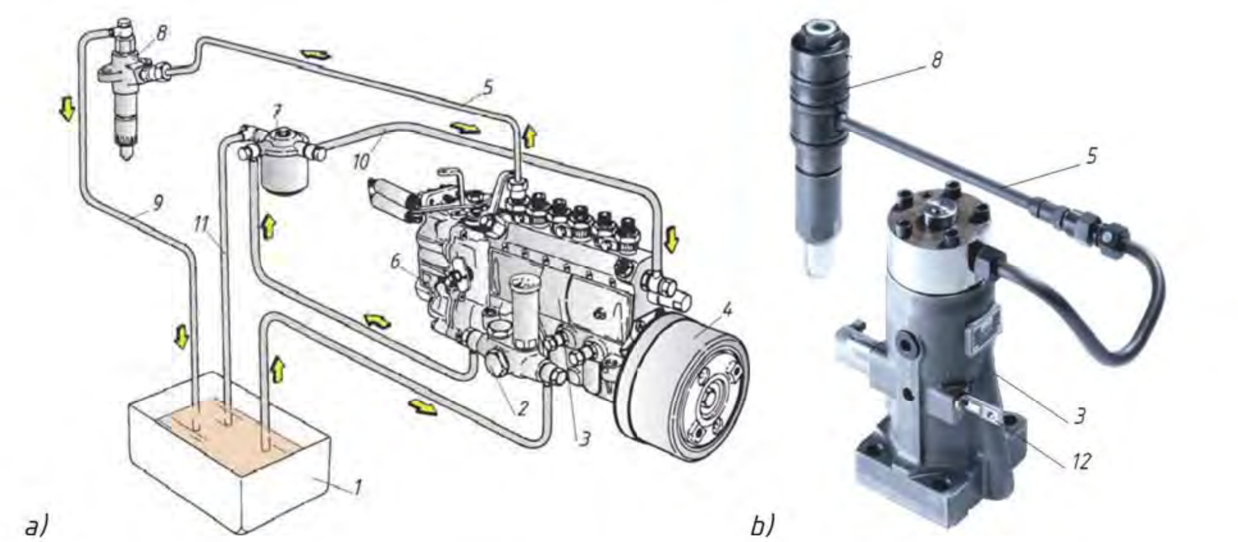


Figure 5.1 – Fuel system with a block-type injection pump (*a*) and with an autonomous injection pump for each working cylinder (*b*). 1 – fuel tank; 2 – low pressure booster pump; 3 – injection pump; 4 – supply advance clutch; 5 – high pressure fuel line; 6 – regulator; 7 – fuel purification filter; 8 – fuel injector; 9 – line for draining fuel leaks from the injector; 10 – fuel drain line from the injection pump; 11 – line for draining unused fuel; 12 – cyclic fuel supply control rail (adapted from [8])

The use of pump-injectors is associated with difficulties in their placement in the cylinder covers and the complexity of the drive, which, as a rule, is carried out from the camshaft through a system of pushers and rods (Fig. 5.2). Currently, pump-injectors are used primarily on high-speed diesel engines of some companies with cylinder diameters up to 300 mm.

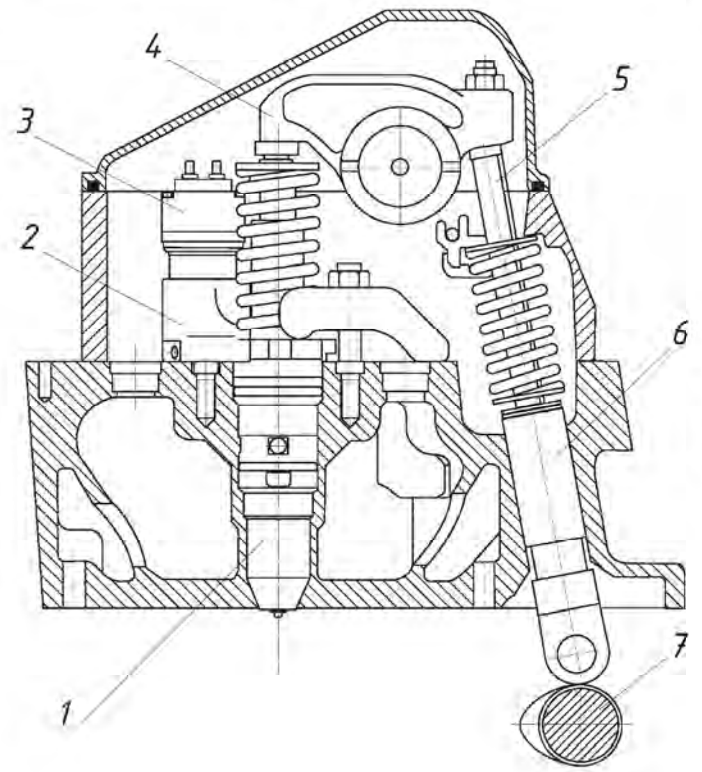


Figure 5.2 – Installation of a pump-injector in the cylinder cover of a high-speed marine diesel engine 3600 series from Caterpillar. 1 – pump-injector; 2 – supply control valve; 3 – solenoid valve; 4 – pump-injector drive rocker arm; 5 – rod; 6 – pusher; 7 – pusher drive cam (adapted from [9])

General diagrams of direct fuel supply systems are shown in Fig. 5.3.

The separate design includes two elements: a high-pressure plunger-type fuel pump and a fuel injector for atomizing fuel, which is connected to the pump by a high-pressure pipeline (Fig. 1.3 *a*). To separate the sub-plunger space and the discharge cavity, a delivery valve is installed between them.

In a number of designs, an intermediate stop is installed between the adjusting bolt and the needle valve spring, in which a groove is made for the fuel supply fitting to the injector to pass through it (Fig. 5.5 *a*, *b*, *c*). The fitting is pressed against the hole on the inside of the wall of the fuel injector housing. The intermediate stop is secured against axial rotation using a locking pin. This solution avoids deformation of the housing under the pressure of the fuel fitting, which can lead to jamming of the injector.

The internal cavity of the chamber for installing the spring is used to collect leaking fuel, which, having leaked along the cylindrical surface of the needle, creates a bath for the spring, ensuring its lubrication and heat removal. This protects the latter from corrosion and reduces dynamic stresses in the turns by 20...25%.

A drain hole for draining leaks into the drainage channel is located in the upper part of the fuel injector to maintain the spring chamber in a constantly filled state.

To prevent leaks from entering the cooling water, special rubber sealing rings are installed on the fuel injector housing, separating the inlet and outlet channels of various media.

The nozzle is the most critical element of the fuel injector design. In Fig. 5.6 shows the designs of some fuel injectors of four-stroke marine diesel engines.

High temperatures operating in the engine combustion chamber can lead to overheating of the nozzle, which can result in jamming of the needle valve, the guide rod of which, together with the guide hole in the nozzle housing, forms a precision pair. As a result of the needle hanging, the nozzle holes become coked. The probability of jamming is especially high for fuel injectors operating on heavy fuels, which are supplied to the fuel injector at a temperature of 100...140°C. Overheating the nozzle results in the locking surfaces' hardness decreasing, their wear increasing, the size of gaps in precision connections changes, and their tightness decreasing. All this is progressive in nature, as it leads to deterioration in the operating process conditions in the engine. The maximum permissible temperature of the fuel nozzle tips should usually not exceed 220...240°C, higher values lead to a rapid decrease in their working capacity.

To prevent overheating, cavities are provided in the fuel injector housing for supplying cooling water (Fig. 5.6 *c*, *d*, *e*) or oil (Fig. 5.6 *b*). These same cavities, as well as the internal channels for supplying coolant, make it possible to maintain the temperature regime of the injector when the engine is not running and is in hot reserve.

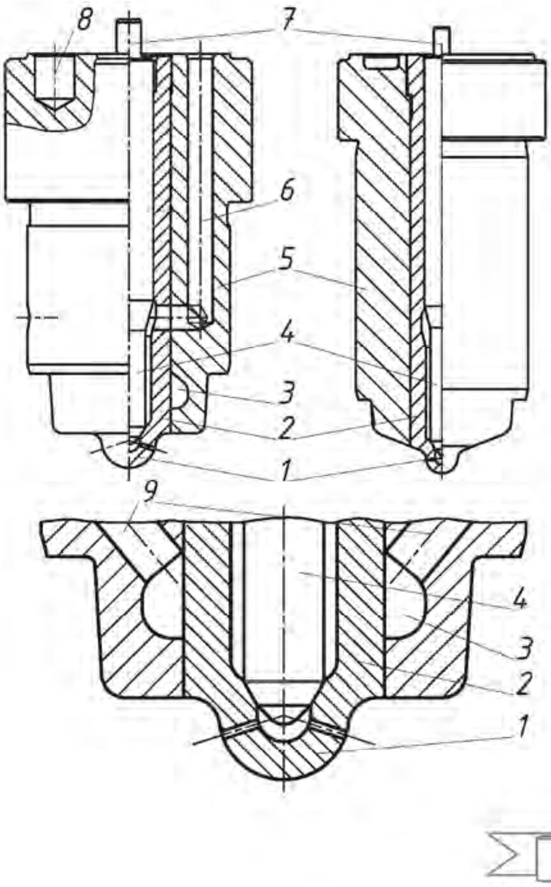


Figure 5.6 – Jet type nozzles: 1 – tip of the nozzle with holes; 2 – guide insert of the needle valve; 3 – cavity for coolant; 4 – needle valve; 5 – spray housing; 6 – channel for supplying fuel to the cavity of the nozzles; 7 – needle valve shank; 8 – hole for the mounting pin; 9 – coolant supply channel (adapted from [15])

The main feature of fuel supply in engines equipped with accumulator injection systems is the presence of constant pressure in the accumulator, which limits the possibility of its sudden release in front of the injector. In a accumulator injection systems, the pressure in the space under the needle will drop only due to the flow of fuel through the fuel injector holes. As the pressure in the supraglottic space drops, the needle will lower, thereby increasing the hydraulic resistance in the gap between the shut-off cone and the needle valve seat. As the gap decreases, the rate of fuel flow from the fuel injector openings will decrease and at the end of injection may not be sufficient to flush the remaining fuel from the fuel nozzle tip. The resulting drop will begin to coke under the influence of high temperatures, and gradually the tip will stop working.

If there is a helical groove (Fig. 5.7 *c*), which has a sufficiently long length, the injection process will proceed as follows: when a relatively large portion of fuel is supplied to the injector from the control unit, the hydraulic resistance of the groove is too large to significantly affect the leakage of fuel from the fuel injectors flange space (the throttling effect of the groove is very large). But when the needle lands, when the fuel supply has stopped, at some point the resistance of the groove will become less than the resistance in the gap between the needle valve and its seat. In this case, the fuel from the space above the needle will follow the path of least resistance, that is, through the helical groove into the spring chamber and then to the drain, and the needle valve will quickly close, ensuring a sharp cutoff of injection.

Another important feature of fuel supply in low-speed diesel engines is the need to maintain the thermal regime of all elements of the fuel equipment to ensure the specified viscosity of heavy fuel. This is especially true when the engine is stopped, since a decrease in fuel temperature can lead to an unacceptable increase in viscosity, at which operation of the fuel system will be impossible. In early designs, this problem was solved by switching the engine to low-viscosity fuel before stopping, which filled the fuel supply system and ensured reliable starting of the engine from a cold state. Today, this procedure is performed only when it is necessary to stop not only the engine itself, but also all its systems.

In some fuel systems, to maintain the temperature, local heaters are installed, so-called satellites, which heat the elements of the fuel equipment due to the heat of the water vapor supplied to them.

Currently, fuel systems with constant circulation of heated fuel are widely used. In this case, the fuel circulates not only while the engine is stopped, but also during the periods between injections. This ensures not only maintaining the specified fuel viscosity, but also cooling the injectors.

To ensure constant circulation, a number of changes have been made to the design of the fuel system elements, the main of which are the replacement of the high-pressure fuel injection valve with a filler valve and the installation of circulation valves in the injectors. The heated fuel, supplied by an electrically driven booster pump during the periods between injections, enters the injection pump space above the plunger through the open delivery valve, from which it enters the injector through a high-pressure pipeline. Next, the fuel enters the cooling cavity of the fuel injector through the open circulation valve, from which it is drained back into the supply tank through the drainage channel.

The operating diagram of a MAN low-speed engine fuel injector equipped with a circulation valve is shown in Fig. 5.9.

A needle-type circulation valve installed at the top separates the injector high-pressure cavity from the fuel supply line. In the absence of supply, fuel enters the cavity of the circulation valve with a pressure of about 1.0 MPa created by the booster pump. This pressure is not enough to open the valve, overcoming the force of the spring loading it. There is a small drain hole in the upper valve guide through which fuel flows from the circulation valve cavity into the cooling cavity of the injector. Next, through the drain fitting, the fuel is discharged back into the supply tank (Fig. 5.9 *a*).

At the beginning of the injection stroke of the injection pump, the drainage hole is unable to drain all the fuel coming from the injection line. As a result, the pressure in the valve cavity begins to increase, which leads to its opening. When the valve is lifted, the drainage hole on the guide is blocked, and the valve cavity is disconnected from the drain line. From this moment, all the fuel supplied by the injection pump enters through the open circulation valve into the cavity of the fuel

precisely fitted to each other, it is extremely important that when installing the injector into the cylinder cover, no stress is created that could lead to their deformation. For this, special shock absorbers are used, which are a cylinder filled with a set of disc springs (Fig. 5.12).

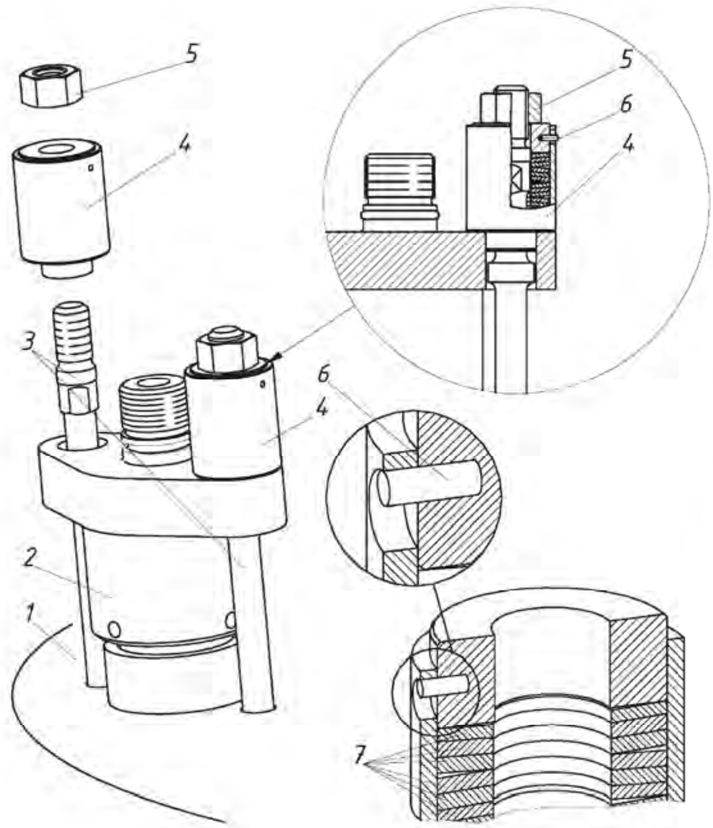


Figure 5.12 – Elastic shock absorber for normalizing the injector tightening value (MC and ME series engines from MAN): 1 – cylinder cover; 2 – fuel injector; 3 – fuel injector mounting studs; 4 – elastic shock absorber; 5 – fuel injector fastening nut; 6 – control pin; 7 – set of disc springs (adapted from [17])

The shock absorbers are put on the mounting studs and rest their bottom on the fuel injector flange. The tightening force from the nuts through the upper washer is transmitted to the fuel injector flange through a set of springs.

The shock absorber housing has a control hole in which a pin is placed, pressed into the upper washer. When properly tightened, the pin occupies a central position in the control hole.

High pressure fuel lines serve to supply fuel from the injection pump to the injectors. To reduce losses in supply pipelines, they try to make them as short as possible. Currently, two types of fuel pipelines, shown in Fig. 5.13.

In the first case, a steel tube is installed between the injection pump and the fuel injector fitting, covered with a protective shell on top (Fig. 5.13 *a*). The casing serves to prevent fuel leakage in the event of damage to the main pipeline. In addition, the space between the high pressure pipe and the protective casing is used to collect and drain leaks from the fuel injector and the connections of the pipeline itself. The ends of the tubes are made in the form of conical thickenings, which are pressed against the wells of the fittings using union nuts. The fuel supply fitting to the injector itself is pressed against the fuel intake well of the injector using an elastic clip. This allows you to avoid excessive pressure in the pipeline in case of fuel injector jamming. The fitting, under the influence of pressure, will overcome the pressing force of the clip, and the fuel supplied by the injection pump will drain into the drainage channel.

The pipeline design shown in Fig. 5.13 *b*, consists of two fittings passing inside the drillings in the cylinder cover. The fittings are pressed against the wells of the fuel receiving channel of the injector and the mating surface of the fuel pump, as well as against each other, using special screw plugs that are screwed into the lug on the cylinder cover. One of the plugs contains an elastic element that protects the high-pressure line from excessive loads. The internal cavity of the tide together with the protective casing forms a leakage collection box. The cavity formed between the drilling in the cylinder cover and the fuel supply fitting to the injector is used to drain leaks.

The rack is connected through a rod system to the speed controller directly or through a hydraulic booster system.

In spool pumps, three main methods are used to regulate the cyclic portion and phases of fuel supply to the working cylinders (Fig. 5.18).

With the first control method (Fig. 5.18 *a*), the start of fuel supply (point A) remains unchanged regardless of the engine speed and load. End of injection (points B_1, B_2 and B_3) changes due to a change in the position of the cut-off edge when the plunger is rotated relative to the bypass holes. This method of regulation is called *end-of-supply regulation.*

In pumps with this type of regulation, the end of the flow at all loads occurs in the area where the plunger rises, where its speed is close to maximum. This ensures high pressure and injection speed throughout the entire fuel supply section, resulting in good fuel atomization quality.

With the second control method (Fig. 5.18 *b*), the beginning of the fuel supply changes (points A_1, A_2 and A_3), and the end of the supply remains unchanged (point B). In these pumps the bevel edge is located at the top plunger when moving upward, the edge, depending on the rotation of the plunger, sooner or later closes the inlet hole in the sleeve, after which its active stroke begins. The end of the active stroke corresponds to the beginning of the cutoff (point B). Thus, in a pump of this type, with a change in the cyclic supply, the fuel injection advance angle simultaneously changes. This method of regulation is called *regulation at the beginning of supply.*

Rotating the plunger leads not only to a reduction in the cyclic portion of fuel, but also to a decrease in the supply advance angle. This leads to the fact that the beginning of the supply is shifted to a section of higher speeds of movement of the plunger (from point A_1 to point A_3), which leads to a more intense increase in pressure at the initial stage of injection and helps to improve the quality of atomization and combustion of fuel. This is especially important when the engine is running at low loads, since as the speed decreases, the injection pressure drops proportionally. Reducing the advance angle and intensifying injection makes it possible to optimize the heat release process while reducing the rotation speed and load on the engine.

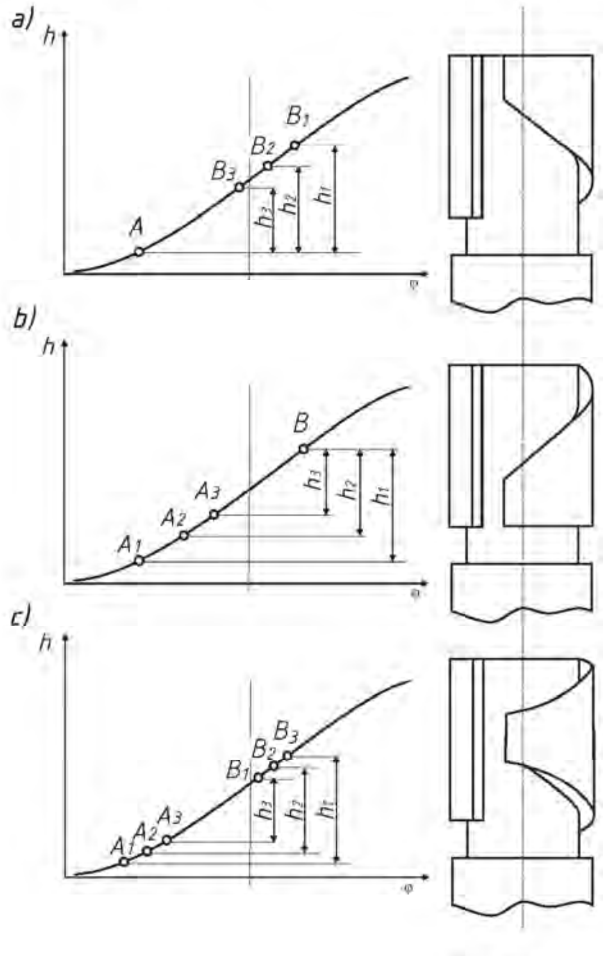


Figure 5.18 – Methods for regulating the cyclic portion and timing of fuel supply to the engine cylinders: *a* – at the end of supply; *b* – at the beginning of the supply; *c* – at the beginning and end of the supply (adapted from [9])

rigid algorithm, which cannot be changed during operation. The need for more flexible regulation may arise when there are significant deviations from the nominal operating modes, when switching to other types of fuel and their mixtures, as well as in a number of other cases. In order to more flexibly regulate the fuel supply process, Burmeister & Wain has developed a spool pump design with a variable fuel injection advance angle (Variable injection timing system hereinafter – VIT). Subsequently, this system was inherited by MAN, which still uses it in its developments to this day.

A general view of the fuel pump with the VIT system, its design and main elements are shown in Fig. 5.22 *a*, *b*, *c*.

In pumps of this type, the advance angle is changed by axial movement of the plunger sleeve relative to the plunger itself. In this case, the position of the cut-off holes relative to the upper edge of the plunger and, consequently, the moment of their overlap relative to the angle of rotation of the engine crankshaft changes.

To use heavy fuels in all engine operating modes, the pumps and injectors are designed in such a way that during parking and during periods between injections, heated fuel circulates in the system, ensuring its heating.

The fuel pump housing has a square base that secures it to the pusher housing. To collect leaks, a special groove is made at the base of the pump, from where leaking fuel flows into a special drain pipe.

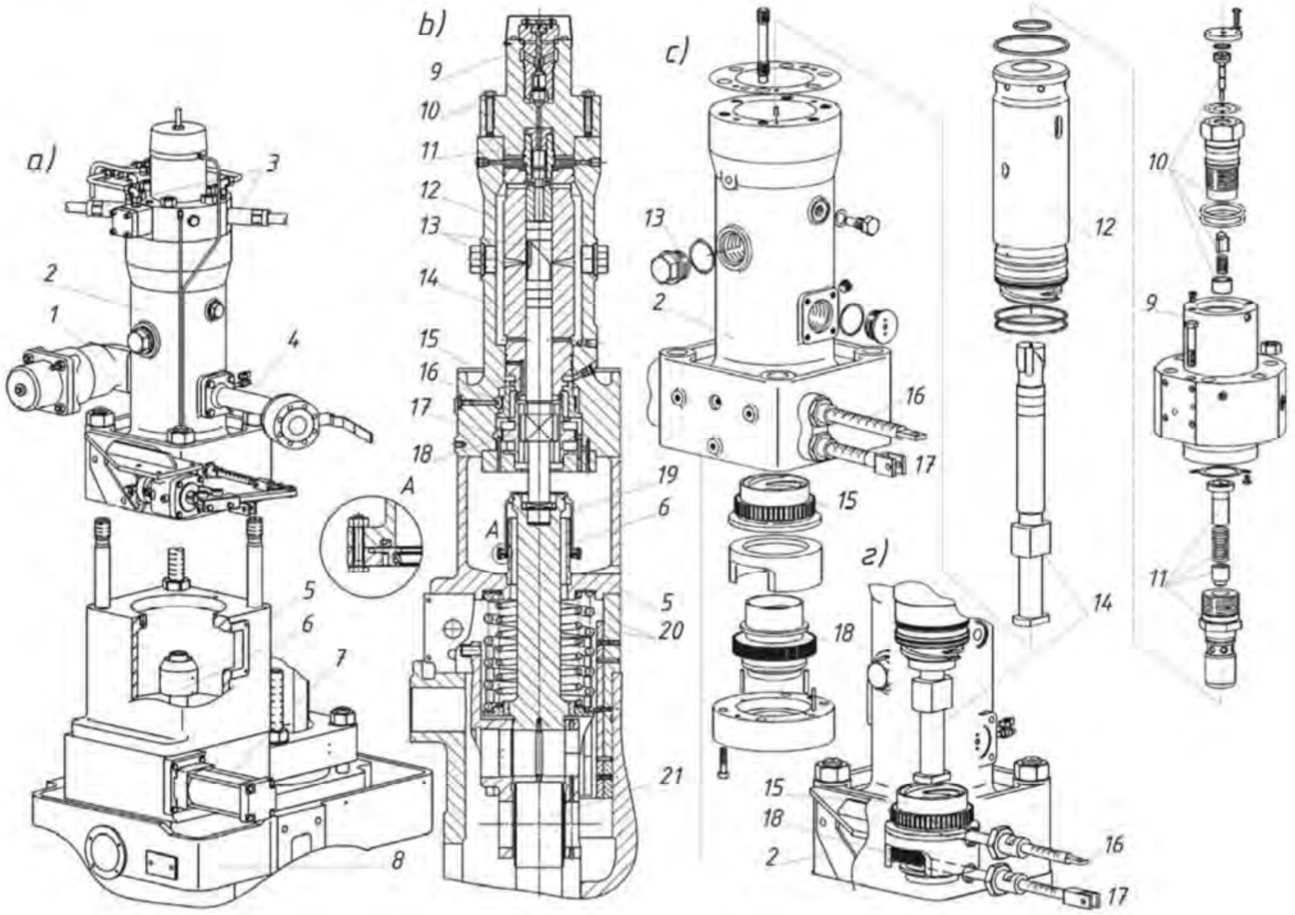


Figure 5.22 – Fuel pump of a low-speed engine of the MC series from MAN with spool control of cyclic supply and fuel injection advance angle (VIT system): 1 – spring pressure damper; 2 – fuel injection pump housing; 3 – high pressure tubes; 4 – fuel supply pipe to the pump; 5 – pusher housing; 6 – pusher; 7 – pneumatic cylinder for driving the reversing mechanism; 8 – camshaft housing; 9 – pump cover; 10 – bypass valve; 11 – suction valve from the upper guide of the plunger bushing; 12 – plunger bushing; 13 – replaceable plugs; 14 – plunger; 15 – rotary sleeve for adjusting the supply advance; 16 – gear rack for adjusting the supply advance angle; 17 – toothed rack for adjusting the cyclic fuel supply; 18 – plunger reversal sleeve; 19 – sliding cuff of the pusher; 20 – return springs; 21 – pusher roller (adapted from [17, 22])

A special cam is used to drive the roller tappet, which ensures optimal fuel supply regardless of the direction of engine rotation. The profile of the drive cam is shown in Fig. 5.28.

The downward movement of the pusher and the pressing of its roller against the cam is ensured by two spiral springs fixed between the pusher and the base of the pump.

The top of the annular groove of the pusher is located inside the pump base and is equipped with a cap. This cap, together with the sealing sleeve, which is hot pressed into the base of the pump, forms a labyrinth that prevents fuel from entering the camshaft lubrication system (Fig. 5.22 *b*).

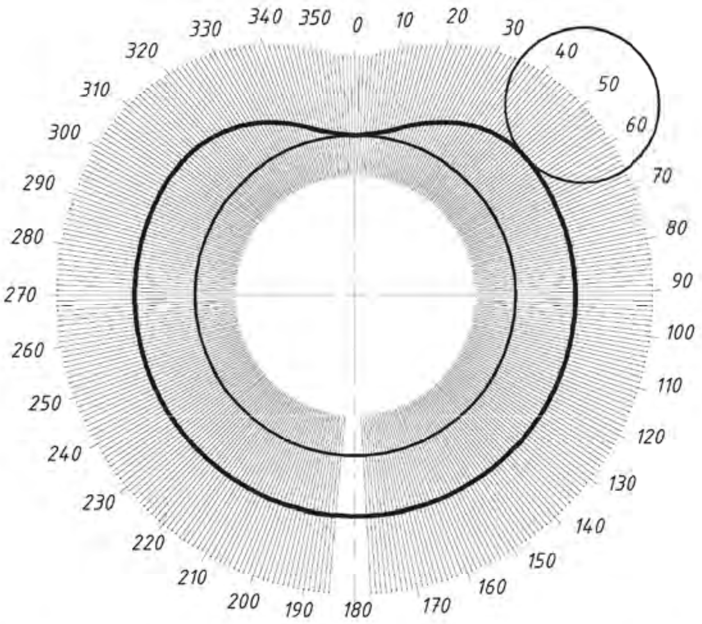


Figure 5.28 – Profile of the fuel cam of diesel engines of the MC series from MAN (adapted from [17])

Each tappet housing has an eccentric-type lifting and locking device that can be used to lift and lock the tappet over the fuel cam. The lifting device is installed on the side of the pusher housing (Fig. 5.29).

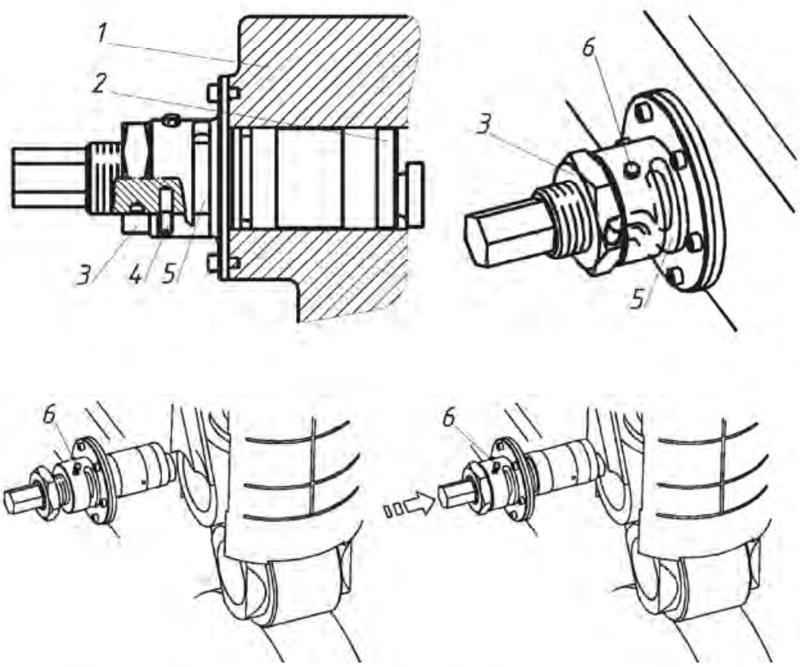


Figure 5.29 – Lifting and fixing device of the fuel injection pump pusher of a MAN MC series diesel engine of eccentric type: 1 – pusher housing; 2 – eccentric lift; 3 – lock nut; 4 – guide pin; 5 – guide groove; 6 – locking screw (adapted from [17, 26])

To raise the pusher, the lock nut on the eccentric shaft is released, and the eccentric protrusion is inserted under the reverse lever by axial movement. Next, by turning the eccentric, the pusher is raised and its position is fixed using a locking screw.

Otherwise, the problem of regulating the moments of the beginning and end of the supply is

the drain cavity through the drain valve. The geometric start of injection is determined by the moment of closing the suction valve, and the geometric end of injection depends on the moment of opening of the drain valve. The opening moments of the valves are determined by the size of the gaps S in the drive mechanism, which are regulated by turning the eccentric shafts (Fig. 5.33).

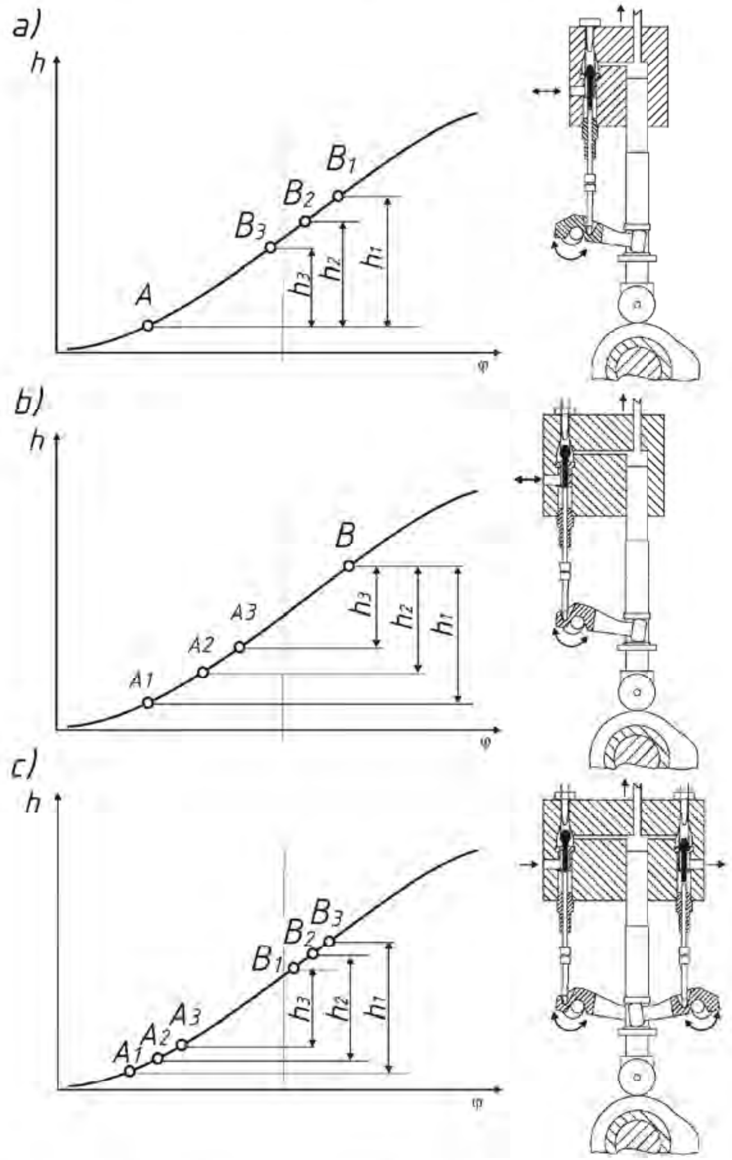


Figure 5.34 – Valve control of fuel supply: *a* – at the end of supply; *b* – at the beginning of the supply; *c* – at the beginning and end of the supply (adapted from [9])

A general view of the pump block, which includes the injection pump, is shown in Fig. 4.31, 5.35, and a cross section of the pump block along the injection pump axis is shown in Fig. 5.36 *a*. The same figure shows the individual elements of the pump design.

The valves are driven through a system of levers and rods from a recess on the bottom of the plunger. The suction valve is actuated through a double-arm lever, and the drain valve is operated through a single-arm lever. The eccentric shafts of the levers are rotated by the engine speed controller, which controls the advance angle of the start of supply, the angle of the end of injection and, consequently, the duration of fuel supply to the cylinder, on which the size of the cyclic portion of fuel directly depends.

The advantages of injection pump with valve regulation include:

- the absence of adjusting edges on the surface of the plunger, which weaken it and create lateral forces that accelerate wear;
- in pumps of this type, the principle of adjusting the moments of the beginning and end of flow is easier to implement than in pumps with spool control. At the same time, they contain fewer precision pairs.

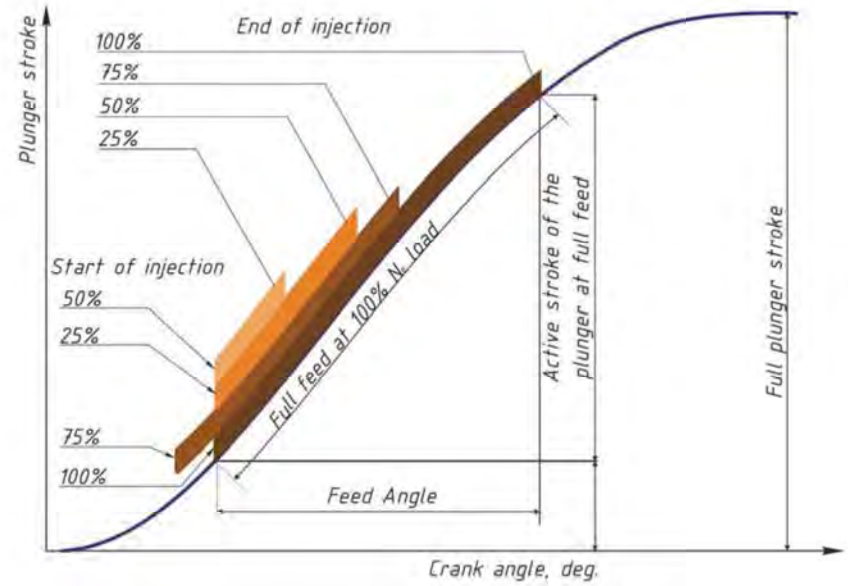


Figure 5.40 – Phases of fuel supply to the injection pump of diesel engines of the RTA series from Wärtsilä (adapted from [33])

5.5.2 Fuel pumps and pump-injectors with solenoid valve actuators

This type of fuel pumps and pump-injectors combines the advantages of valve-controlled pumps and allows the fuel supply process to be controlled within a wide range using controllers based on microprocessor technology. Optimization of the fuel supply law is carried out by a program embedded in the electronic control unit, which receives and processes signals from sensors of rotation speed, load, boost pressure, fuel temperature, etc. Based on the processing of the received signals, the program determines the optimal values for the start and end of fuel supply for a given mode and applies voltage to the control valve drive solenoids. When using pumps with solenoid controlled valves, fuel injection can be carried out through a traditional injector design, which greatly simplifies the implementation of such systems on diesel engines. The only complex and non-traditional element of such injection pumps is the control valve, which is subject to a number of specific requirements. U in high-speed diesel engines, the valve must operate in both directions in a time of no more than 0.1...0.2 ms, and this is possible with electromagnet forces of at least 250 N, and then only with small masses of moving parts.

Figure 5.41 shows a pump-injector with a solenoid actuated control valve, type EI800, which is equipped with Caterpillar 3500 and 3600 series engines.

The operating diagram of the EI800 type pump-injector is shown in Fig. 5.42.

When the plunger moves upward, the space under the plunger is filled. Fuel from the low pressure line enters through the open control valve through the bypass channel into the cavity under the plunger (Fig. 5.42 *a*).

As the plunger moves downward (discharge stroke), the pressure underneath it will not increase as long as the control valve is in the open position.

When voltage is applied from the control unit to the drive solenoid, the valve rises and sits on the seat, as a result of which the cavity under the plunger will be disconnected from the low pressure line. The pressure in the cavity under the plunger begins to increase, and when it reaches 35 MPa, the fuel injector needle, overcoming the force of the spring, rises and fuel injection into the combustion chamber begins (Fig. 5.42 *b*).

When the power to the solenoid is turned off under the action of a spring, the valve opens and the fuel supply to the cylinder stops (Fig. 5.42 *c*), then the plunger idles, and then the whole cycle repeats again.

This system allows you to supply a cyclic portion of fuel both in one injection and to split the cyclic portion into several successive injections.

Cummins used its well-developed scheme using open-type pump-injectors, adapting their design to the electronic engine control system. This fuel system is widely used on high-speed engines, which are used on merchant ships mainly to drive diesel generators.

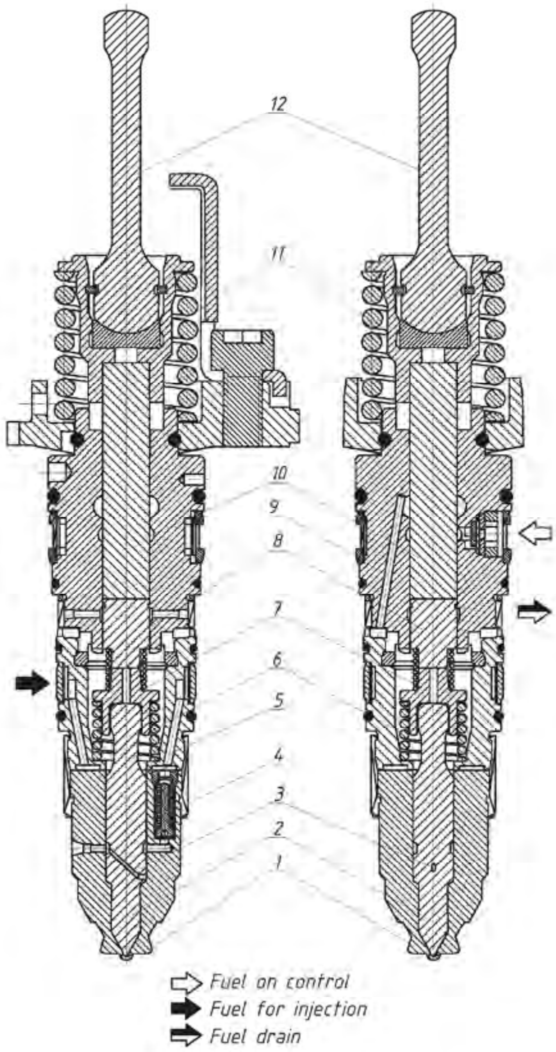


Figure 5.44 – Pump-injector of Cummins engines with an electronic supply control system type ISX: 1 – spray housing; 2 – injection plunger; 3 – drainage channel; 4 – suction valve; 5 – filling channel; 6 – intermediate spring; 7 – pusher of the injection plunger; 8 – intermediate plunger; 9 – leakage channel; 10 – upper plunger; 11 – return spring; 12 – pusher rod (adapted from [35])

The operating procedure of the pump-injector of the ISX system is shown in Fig. 5.46.

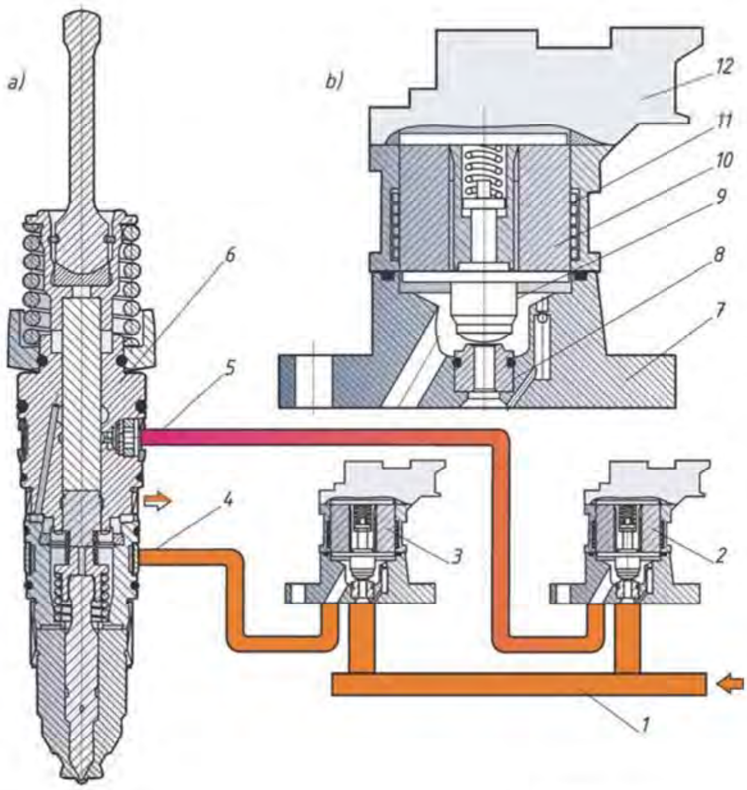


Figure 5.45 – General diagram of the fuel supply control in the Cummins ISX injection system (*a*) and control valve diagram (*b*): 1 – fuel supply line from the fuel pump; 2 – injection control valve; 3 – cyclic supply dosing valve; 4 – fuel supply channel for injection; 5 – channel for supplying fuel to the system for regulating the start of supply; 6 – pump-injector; 7 – valve housing; 8 – valve seat; 9 – shut-off cone; 10 – magnetic core; 11 – solenoid coil; 12 – electrical connector (adapted from [9, 35])

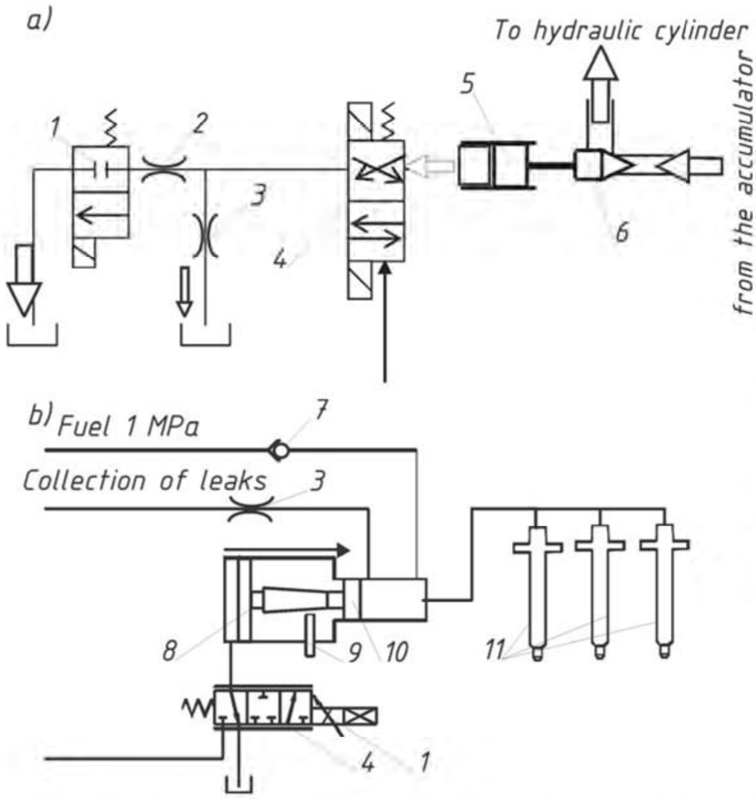


Figure 5.48 – Schematic diagrams of the hydraulic drive of the injection pump of the UEC Eco-Engine series engine from Mitsubishi (*a*) and ME from MAN (*b*): 1 – pilot valve driven by a solenoid; 2 – main throttle; 3 – drain throttle; 4 – main distributor; 5 – main valve drive piston; 6 – main valve; 7 – check valve; 8 – hydraulic piston; 9 – position sensor; 10 – fuel plunger; 11 – injectors (adapted from [9, 37])

The distribution mechanisms of fuel pumps and exhaust valve actuators, as a rule, have an identical design, and in some cases they are unified. The operating principle and general design of spool valves for third generation ME series engines are shown in Fig. 5.49.

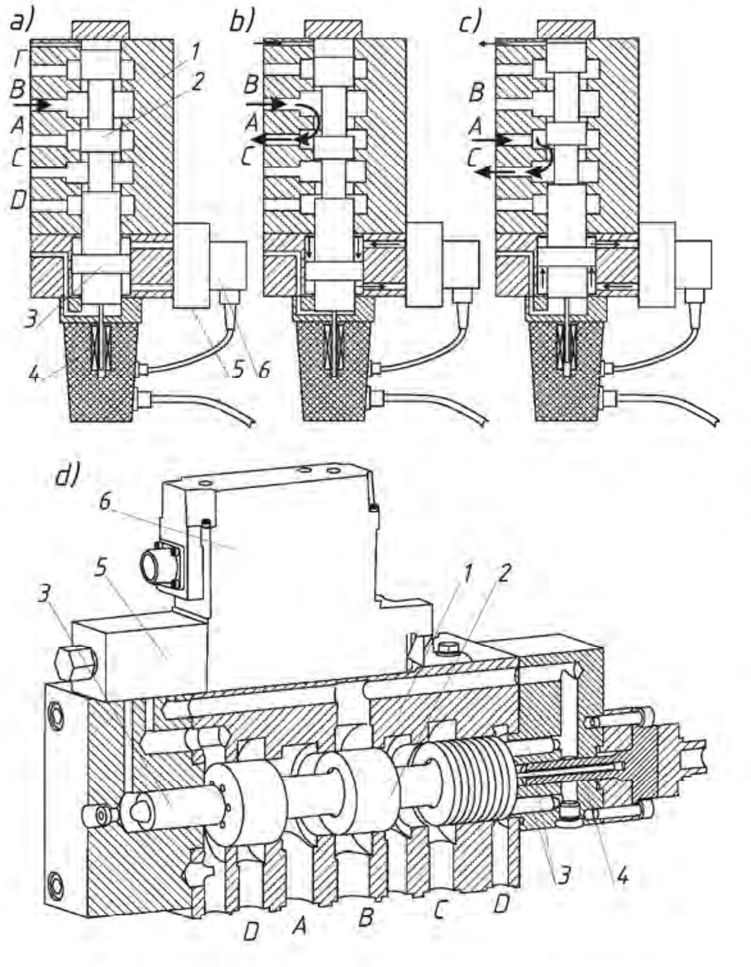


Figure 5.49 – Diagram of the operation of the spool distributor of the injection pump of the ME series engine from MAN (*a*, *b*, *c*) and its general view (*d*): 1 – housing; 2 – main spool; 3 – main spool drive piston; 4 – displacement sensor; 5 – pilot valve; 6 – solenoid; *A* – hydraulic cylinder cavity; *B* – high pressure cavity; *C* – drain cavity; *D* – leakage collection cavity (adapted from [38])

Hydraulic pump-injector (FBIV module)

At the design stage of a new injection system, the company's specialists considered two design options for the FBIV module. The first option (Version *A*) was equipped with a module with direct control from an electronic unit, which is a control spool valve with an electrohydraulic drive and a feedback sensor (Fig. 5.53 *a*). The second option is a simplified version without a separate control valve on each FBIV module (Version *B*) (Fig. 5.53 *b*).

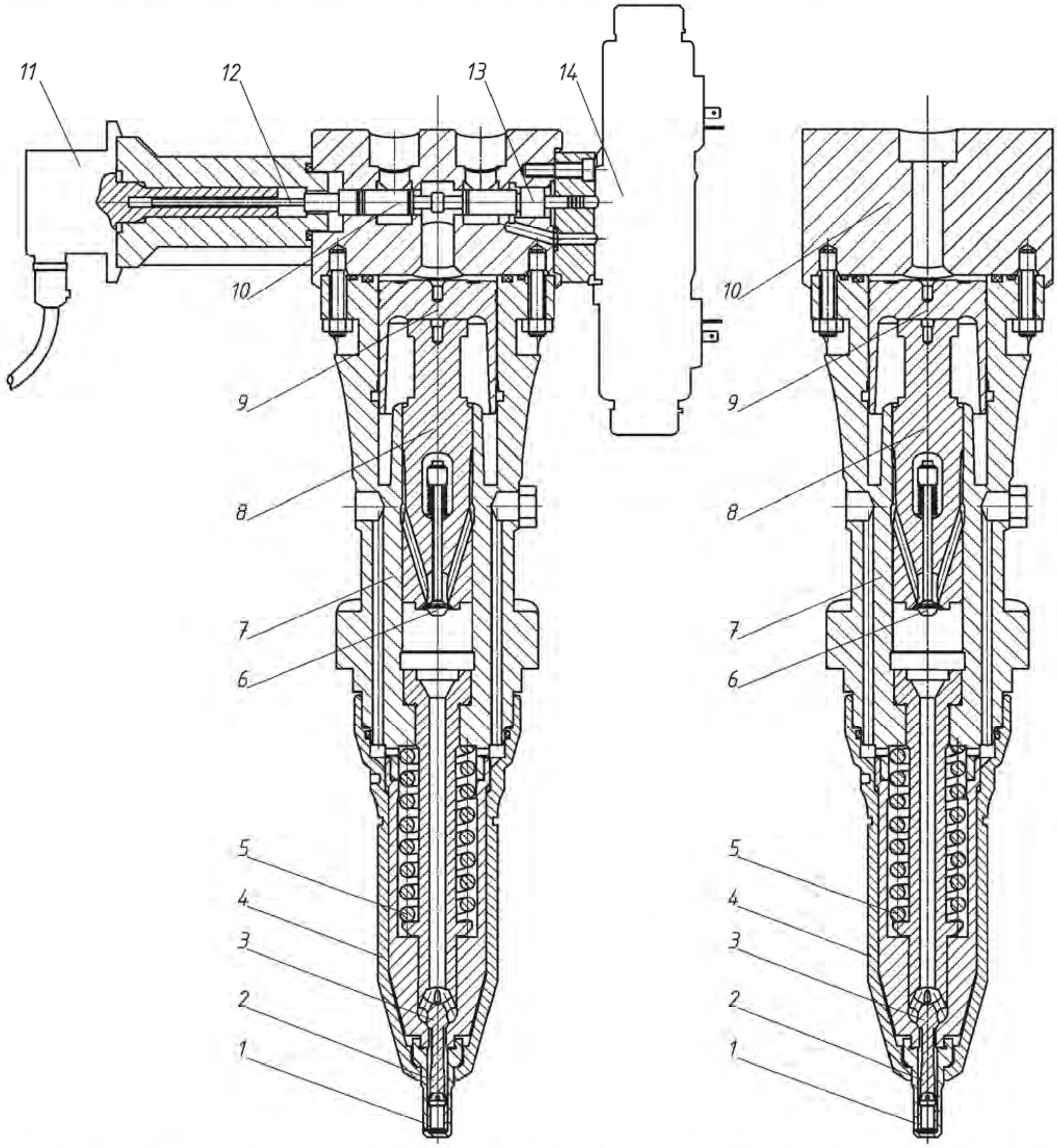


Figure 5.53 – Hydraulic pump-injectors of low-speed engines from MAN series Mk 10 version *A* (*a*) and in version and *B* (*b*): 1 – spray tip; 2 – spool valve; 3 – needle valve; 4 – cap; 5 – needle valve spring; 6 – suction valve; 7 – pump-injector housing; 8 – plunger; 9 – hydraulic piston; 10 – control spool; 11 – control spool position sensor; 12 – measuring rod; 13 – hydraulic piston of the control spool drive; 14 – solenoid control valve (adapted from [42, 43])

The operating procedure of the pump-injector version *A* is shown in Fig. 5.54.

Oil is supplied to the cavity of the hydraulic cylinder through a poppet valve, which can have a horizontal (Fig. 5.58 *a*) or vertical (Fig. 5.58 *b*) arrangement. The valves are opened by a solenoid switched from the electronic control unit, and closed by a return spring. The valve is pressed against the seat under the influence of pressure in the oil line. In the open state, the valve communicates the high-pressure cavity with the cavity of the hydraulic cylinder, and in the closed state, with its unloading device, it communicates with the cavity of the hydraulic cylinder with the drain line. The unloading device can be made in the form of a cylindrical spool (Fig. 5.58 *a*) or a shut-off cone (Fig. 5.58 *b*).

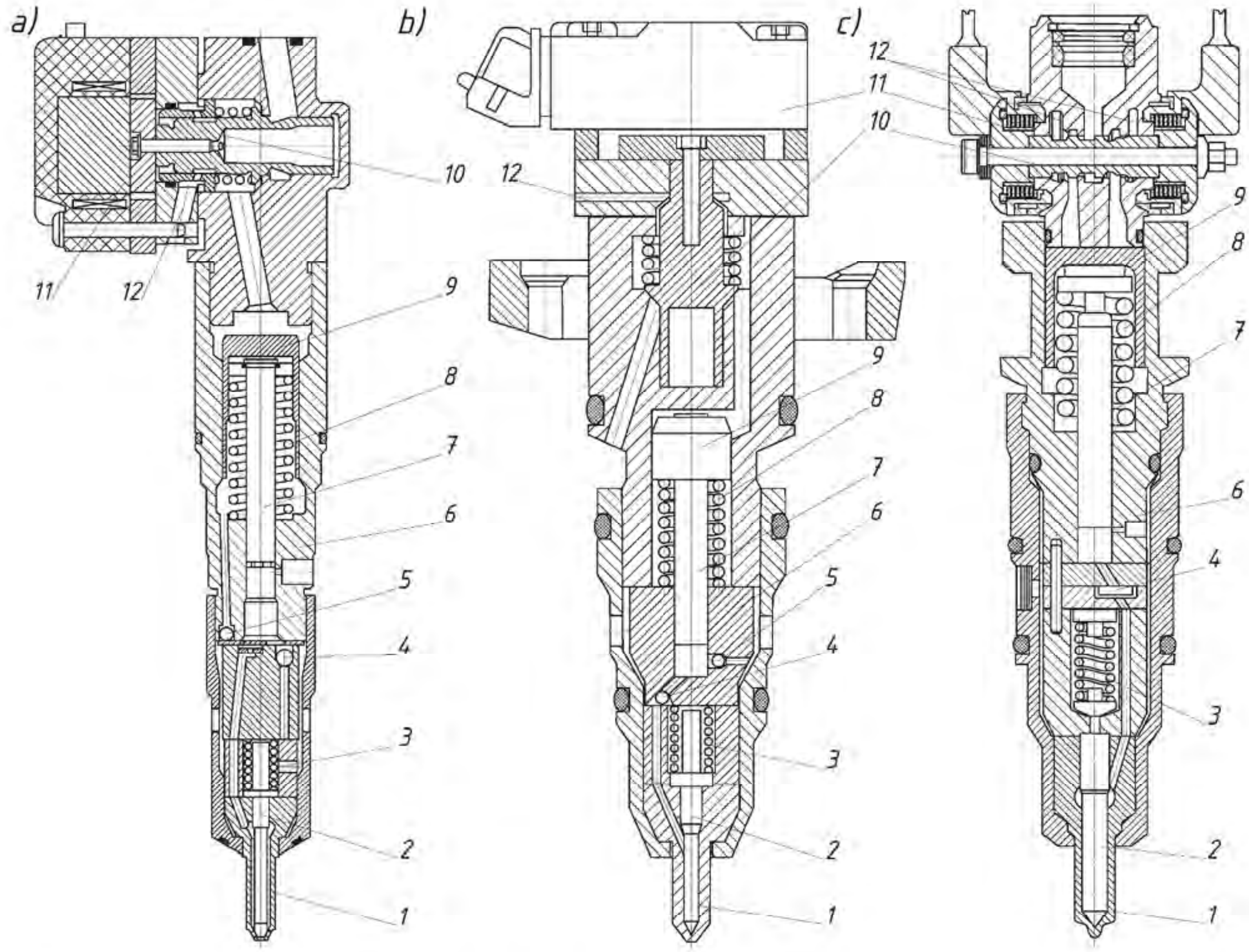


Figure 5.58 – Pump-injectors with hydraulic drive and electromagnetic supply control (HEUI): *a* – first generation with a side-mounted control valve; *b* – first generation with a central location of the control valve; *c* – second generation G2 with double bistable control valve: 1 – spray housing; 2 – needle valve of the nozzle; 3 – needle valve spring; 4 – suction valve; 5 – check valve; 6 – plunger bushing; 7 – plunger; 8 – return spring; 9 – hydraulic piston of the plunger drive; 10 – control valve; 11 – solenoid of the electromagnetic drive of the control valve; 12 – drain line (adapted from [44, 45, 46])

Experience in operating these pump-injectors has shown that their use is only possible on relatively small engines with limited cyclic supply volumes. To ensure the necessary speed when increasing cyclic supplys, it became necessary to increase the size of the control valve, which would lead to the need to apply significant forces to open it. Creating the required force using a solenoid turned out to be quite problematic, since it would require large currents through the coil, which, given the limited size of the solenoid, would lead to its rapid overheating and failure.

Therefore, in their further developments, manufacturing companies switched to the use of bistable spool valves.

The second generation pump-injector type G2 with a double bistable control valve is shown in Fig. 5.58 *c*.

of the plunger drive. In this case, the loading rod of the drain valve is free from the stop and the oil from the cavity of the hydraulic cylinder freely flows into the drain line of the power circuit. The hydraulic piston, under the action of the return spring, moves to its uppermost position, and a new portion of fuel enters the sub-plunger space through the suction valve.

When voltage is applied from the electronic control unit to the solenoid coil, the control valve moves to its highest position, connecting the cavity of the pump-injector control circuit with the drain line. As a result of the pressure drop in the circuit, the hydraulic locking system of the nozzle needle valve is unloaded.

In this case, the spool valve, under the influence of oil pressure in the power circuit, moves down, connecting the cavity of the hydraulic cylinder with the power circuit (Fig. 5.61 *b*). As it descends, the spool valve acts on the drain valve rod, disconnecting the cavity of the hydraulic cylinder and the drain line.

The oil, acting on the piston, moves it along with the plunger down. Since the plunger area is six times smaller than the hydraulic drive piston area, the pressure in the fuel cavity increases to 170...180 MPa. When the needle valve opening pressure is reached, fuel injection begins into the engine combustion chamber (Fig. 5.61 *c*).

After removing the voltage from the solenoid coil, the control valve moves upward under the action of the return spring, connecting the cavity of the control circuit with the oil line. The pressure in the cavity increases sharply, the oil acts on the loading piston of the needle valve of the nozzle, causing an abrupt stop in flow.

Under the action of the return spring and oil pressure on the lower end of the spool valve, the latter moves upward, disconnecting the cavity of the power circuit and the oil line. In this case, the drain valve rod of the hydraulic cylinder is released, and the oil displaced by the piston flows into the drain line (Fig. 5.61 *c*). The sub-plunger space is filled with a new portion of fuel.

The use of relatively small volumes of oil to drive pump-injectors and two-stage control made it possible to reduce the size of the control valve, which together led to an increase in the speed of the system as a whole. This pump-injector is capable of providing not only standard fuel supply laws, but also some special ones aimed at improving the efficient or environmental characteristics of the engine.

At the same time, the complex hydraulic circuit of pump-injectors does not allow achieving uniform performance in their production. In this regard, the company determines correction characteristics for each unit based on test results, which are supplied with the pump-injector in the form of a software algorithm. This program, in the form of an accompanying file, is installed on the microprocessor control unit when replacing the pump-injector. Subsequently, when generating control signals, the control unit takes into account the individual characteristics of each unit.

Control oil pressure regulator. On small engines, to maintain a given pressure in the control oil system, an electrohydraulic pressure regulator, switched from the electronic control unit, is installed directly in the oil line. The general view of the regulator is shown in Fig. 5.62.

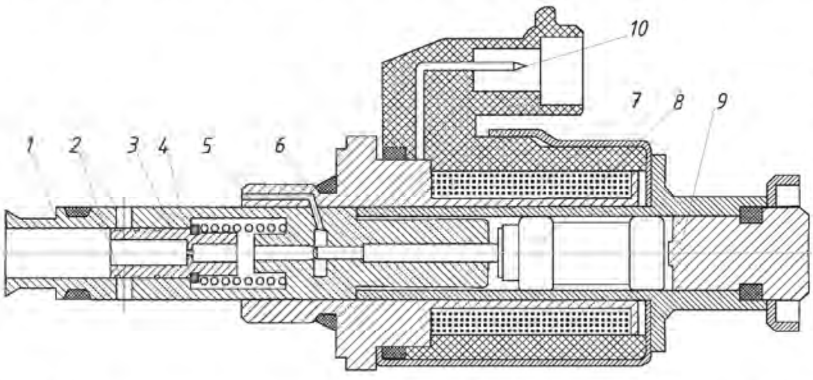


Figure 5.62 – Control oil pressure control valve: 1 – valve housing; 2 – oil drain holes; 3 – spool piston; 4 – throttle hole in the spool piston; 5 – line for draining oil from the differential cavity; 6 – shut-off cone of the system for releasing pressure from the differential cavity; 7 – solenoid coil; 8 – anchor; 9 – anchor stop; 10 – terminal for connecting to the electronic control unit (adapted from [46])

The main regulating housing is the spool piston, which directly interacts with the oil in the high-pressure line. With its edges, the piston blocks the holes for discharging oil into the low pressure line. The piston itself has a calibrated drainage channel, through which the high-pressure cavity

From Fig. 5.65 it can be seen that with traditional fuel injection pumps, the injection pressure decreases in proportion to the decrease in the rotational speed of the fuel cam, and decreases significantly with a decrease in the cyclic fuel supply.

In accumulator systems, injection pressure can be maintained constant regardless of these factors. Moreover, injection pressure and fuel delivery phases are controlled independently of each other. Fuel injection begins at maximum pressure, which contributes to high-quality atomization of the first portions of fuel entering the combustion chamber and faster pre-flame processes.

The use of electronic control in conjunction with an accumulator injection system provides a number of significant advantages over traditional fuel supply systems:

- ensuring flexible regulation of the law of supply and quantity of injected fuel in accordance with the specified load-speed operating conditions of the diesel engine;
- ensuring the necessary unevenness of fuel supply to the cylinders, including taking into account the individual characteristics of individual elements of the fuel supply system, their technical condition and the condition of the elements of the engine itself;
- shutdown of cylinders at partial loads;
- combining the control function with the engine diagnostic function through a system of control sensors, providing emergency engine protection.

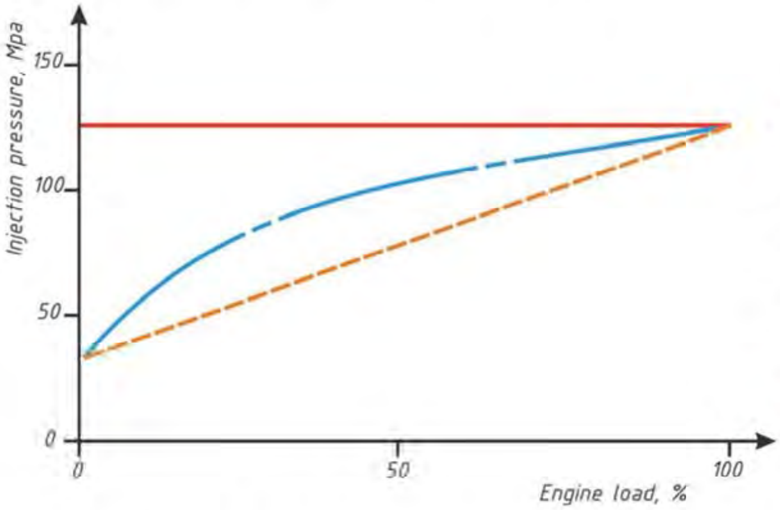


Figure 5.65 – Dependence of fuel injection pressure on engine load for various fuel supply systems:
– – – – direct-acting system when operating according to the screw characteristic (depending on the rotation speed);
—— – —— – direct action system when working according to the load characteristic (depending on the value of the cyclic supply);
——— – accumulator injection system (adapted from [9])

Of the variety of circuit solutions for accumulator injection systems on marine diesel engines, the most widely used are systems with pressure accumulators, made in the form of separate elements of large and small capacity.

5.7.1 Accumulator injection systems for low-speed diesel engines of the RT-flex series from Wärtsilä

Currently, the accumulator fuel injection system is used on low-speed diesel engines of the RT-flex series, produced by Wärtsilä, which inherited the developments of Sulzer, which released the first engine of this series in 2001. The system was designed to operate on heavy residual fuel HFO in accordance with ISO specification DIN 8217 (viscosity up to 700 cSt at 50°C) at temperatures up to 150°C.

The fuel system of RT-flex diesel engines is one of the components of the integrated engine management system, which also includes subsystems for controlling gas distribution, cylinder lubrication and starting air. The general diagram of the diesel control system of the RT-flex series is shown in Fig. 5.66.

The control functions of all subsystems are assigned to the microprocessor control unit WECS-9500 (Wärtsilä Engine Control System) (and its modifications), which is installed on each engine cylinder. All blocks are interconnected by a single system bus; if one block fails, its functions are automatically distributed among others. Thus, the failure of one block does not lead to failure of the cylinder.

channels through which fuel leaks are diverted to the leakage collection tank. In addition to the connecting fittings, safety valves adjusted to a pressure of 125 MPa and fittings for connecting measuring instruments are installed on the accumulator housing (Fig. 5.71 *b*).

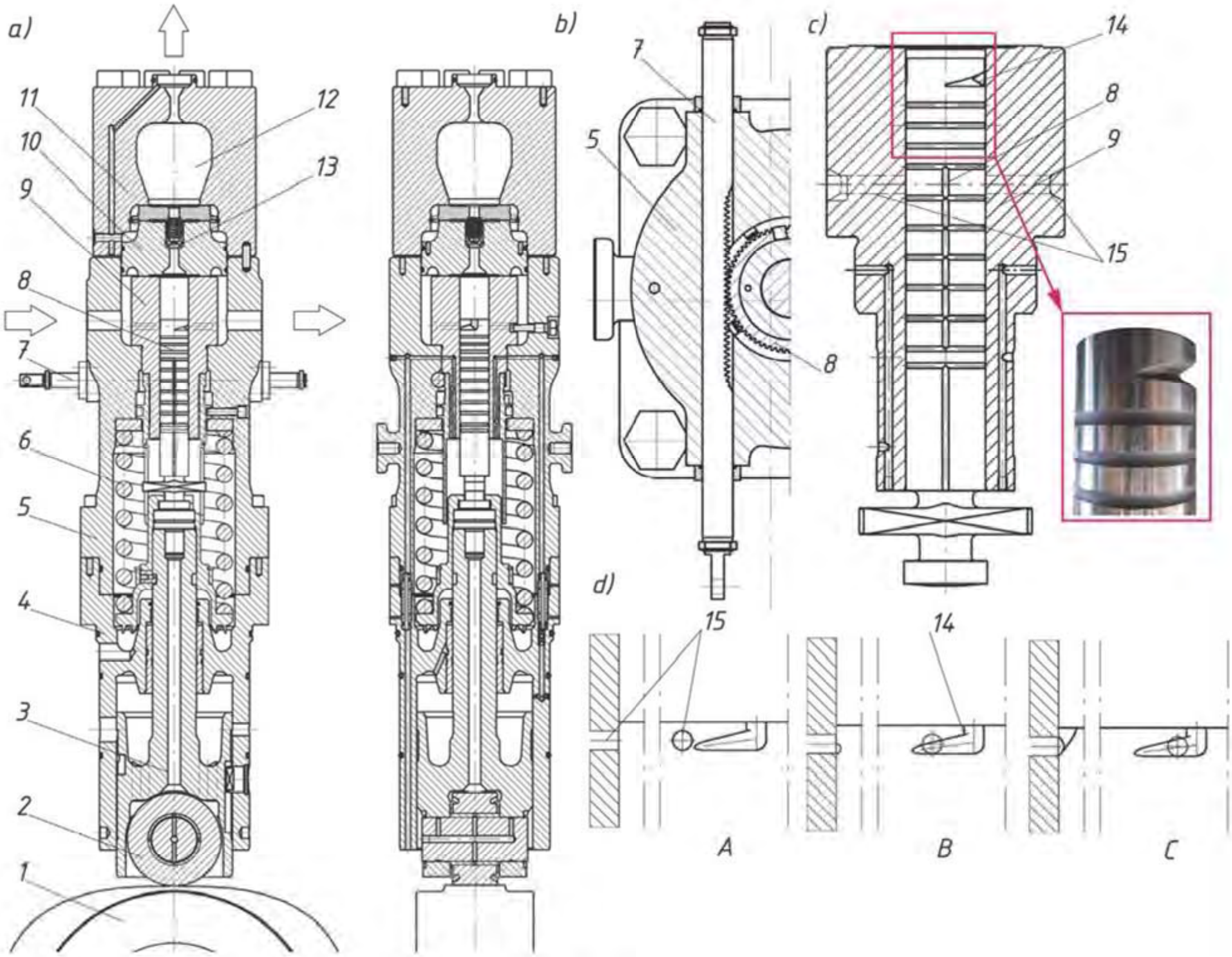


Figure 5.70 – High-pressure section of the RT-flex 96C diesel engine (*a*), rack-and-pinion plunger rotation mechanism (*b*), plunger pair (*c*) and uel pump section delivery control scheme (*d*). 1 – drive cam; 2 – pusher roller; 3 – pusher; 4 – pusher housing; 5 – high pressure section housing; 6 – return spring; 7 – rack; 8 – plunger; 9 – plunger bushing; 10 – check valve housing; 11 – cover; 12 – buffer cavity; 13 – check valve; 14 – throttling groove; 15 – filling hole. *A* – zero supply; *B* – partial supply; *C* – full supply (adapted from [51, 52])

At the ends of the accumulator housing there are covers, to one of which the pressure limiter regulator is attached, and to the other – the connecting fittings of the high-pressure pipelines for supplying fuel to the main accumulator.

In order to ensure greater reliability, two parallel high-pressure lines are laid between the intermediate and main accumulators with shut-off valves installed on them, which allows them to be put into operation both together and separately. The capacity of each line allows the engine to operate at full power.

Between the housing and the covers, special lip seals made of polymer materials are installed, which are pressed against the end surfaces of the cover and housing by the force of fuel pressure.

On engines with a small cylinder diameter, an intermediate accumulator may not be installed, and the fuel goes directly from the high-pressure sections to the main accumulator.

The pressure limiter regulator serves to maintain a constant high pressure in the fuel accumulation system.

The pressure regulation limit is 50...105 MPa, but the recommended operating range is 60...80 MPa.

The injection control unit (ICU) is the most critical element of the fuel system of RT-flex series engines.

Depending on the size of the working cylinder, engines of this series are equipped with two or three fuel injectors per cylinder. Accordingly, the control unit consists of two or three control modules.

Each module includes two high-speed valves – a paired valve for controlling the fuel supply to the injector with a hydraulic drive, and a spool valve with an electromagnetic drive that controls this drive.

The hydraulic diagram of the injection control module is shown in Fig. 5.76.

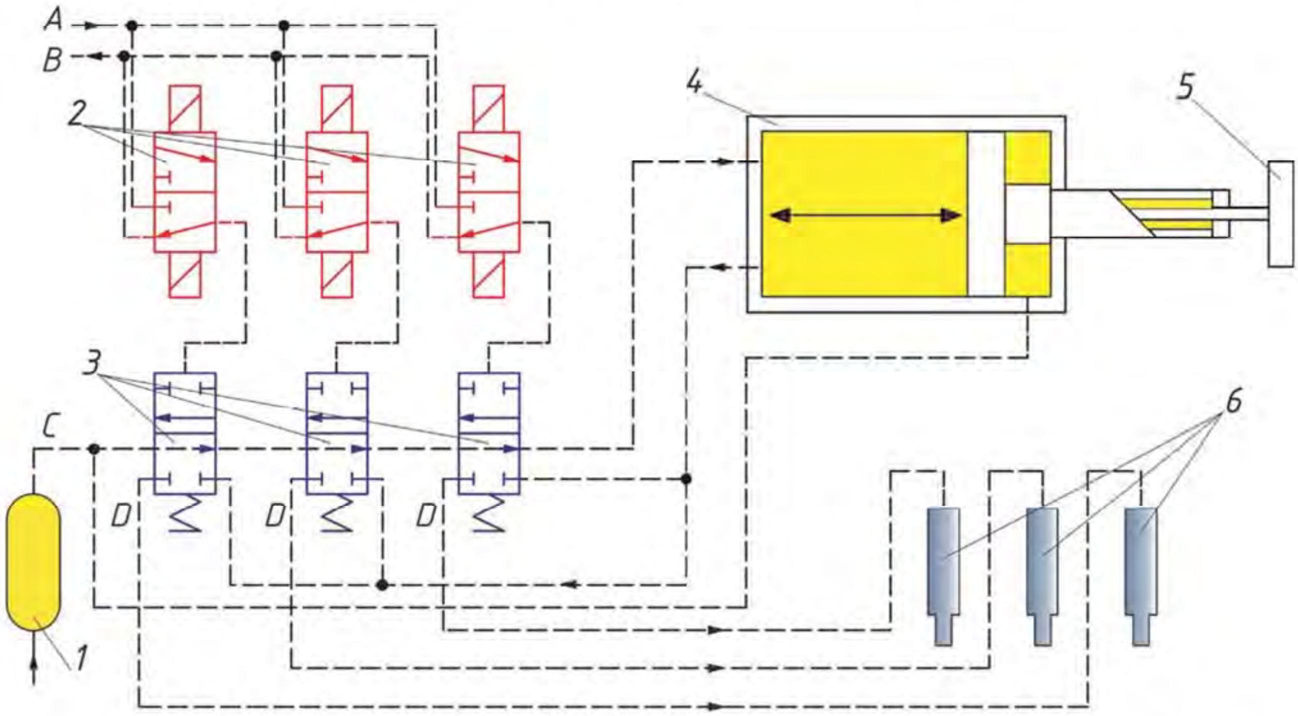


Figure 5.76 – Hydraulic diagram of the injection control module: 1 – accumulator; 2 – spool valve with electromagnetic drive; 3 – valve for controlling the fuel supply to the injector; 4 – measuring cylinder; 5 – fuel consumption sensor; 6 – fuel injectors. *A* – control line; *B* – drain line; *C* – high pressure fuel supply; *D* – fuel to injectors (adapted from [55])

The electric valve is controlled by the WECS-9500 microprocessor module.

For the correct selection of injection parameters, there is feedback between the control unit and the microprocessor module, which is implemented through a sensor for monitoring the amount of injected fuel.

The principle of determining fuel consumption is based on recording the movement of a special measuring piston located in the control unit housing.

The operating principle of the injection control module is shown in Fig. 5.77.

In the absence of a signal from the electronic control unit, fuel, under high pressure, flows from the accumulator into the cavity of the measuring cylinder. To the cavity, on the side of the measuring rod, fuel flows directly from the accumulator, and into the cavity free from the rod – through the open spool valve.

Thus, the same pressure acts on the piston on both sides, however, given that the area of the piston on the side of the measuring rod is smaller by the cross-section of the rod itself, a force of greater magnitude acts on the free side of the piston. As a result of the force difference, the piston moves in the direction of the rod and moves the sensor screen to its extreme position.

In addition, from the cavity of the measuring cylinder, fuel enters the cavity above the main supply control valve, but since the valve is closed at this moment, fuel is not injected into the cylinder.

while simultaneously reducing the number of elements included in the system. In this case, the element base was partially used, developed for first-generation systems, and partially for four-stroke medium-speed engines. The system was initially developed for low-speed engines with a cylinder diameter of up to 50 cm. The first 6RT-flex35 engine equipped with the new system was released in 2011. These developments were subsequently inherited by Winterthur Gas and Diesel (WinGD), which became the successor to Wärtsilä in the low-speed marine engine sector. Based on second-generation technology, WinGD launched the production of a new line of WX series engines with cylinder diameters from 35 to 72 cm.

MAN Diesel & Turbo SE developed its own version of Common Rail fuel systems for low-speed engines with a cylinder diameter of up to 50 cm. The first commercial engine, the 6S35ME-C9-CR, equipped with this system, was built by subsidiary MOL Mitsui O.S.K. Kinkai Ltd. in 2018.

Common rail engines from both manufacturers are designed to operate on liquid fuels in accordance with ISO 8217 DMA, DMZ and DMB, as well as on HFO up to 700 cSt (in accordance with ISO-F-RMK 700). It is also possible to create on their basis engines running on alternative fuels, which we discussed earlier. To a large extent, the systems are similar both in circuit and elemental solutions, for this reason it makes sense to consider them together, focusing only on the differences.

The fuel pressure accumulator in both systems is a single-wall pipe into which fuel is pumped by several high-pressure fuel pumps. The number of installed pumps depends on the number of working cylinders, but not less than two, to ensure that the engine can operate if one of the pumps fails. Small engines (35...50 cm) use multi-section fuel pumps manufactured by L'Orange with throttle performance regulation, which have proven themselves on medium-speed engines (will be discussed in the following chapters). They are distinguished by their reliability, small size and well-established production and service, which significantly reduce the cost of the fuel system. The pumps are designed to create pressures of 160 MPa, while for low-speed engines the operating pressure is limited to 120 MPa. To obtain the specified pressures, the pump rotation speed must be within the range of 2300...3000 min^{-1}. To obtain a given frequency, WinGD engines use a drive from the crankshaft with an overdrive (Fig. 5.81), while MAN uses an independent electric drive.

The electric drive of the pumps is carried out from the on-board network through a frequency regulator, which makes it possible to coordinate the operating modes of the pumps with the operating mode of the engine. The amount of fuel supplied by the pump is controlled by a control unit (throttle valve) installed on the suction side.

Fuel is supplied to the pumps through automatic backwash filters installed in series, designed for a cleaning fineness of 50 and 6 μm.

Lubricating oil is supplied to each pump by taking it from the engine's circulating lubrication system. If higher pressure is required to lubricate the pumps, additional oil pumps are installed.

In large-size engines (52-72 cm) from WinGD, pump sections designed for the first generation RT-flex systems, the design and operating principle of which we discussed earlier, are used to supply fuel to the accumulator.

Each high-pressure pump is connected to the fuel accumulator by separate pipes with double walls through a check valve installed in a special unified housing directly on the fuel accumulator (Fig. 5.82). This is a normally open valve, which is designed to block reverse flow from the accumulator in the event of a leak from the fuel supply pipe from the high pressure pump (Fig. 5.83). During normal operation, fuel pressure opens the valve, overcoming the force of the spring, allowing fuel to flow into the accumulator. This ensures that the fuel system can continue to operate if one of the pipes is damaged or one of the fuel pumps fails.

Combination pressure control valves are installed on the end caps of the accumulators (MAN installs two valves that duplicate each other) which act as mechanical safety valves and as controlled fuel pressure regulators in the accumulator. These same valves are used to remotely relieve pressure during an emergency engine shutdown and to circulate heated fuel through the system when the engine is stopped.

The valve design is similar to those used in the first generation RT-flex engines. The opening

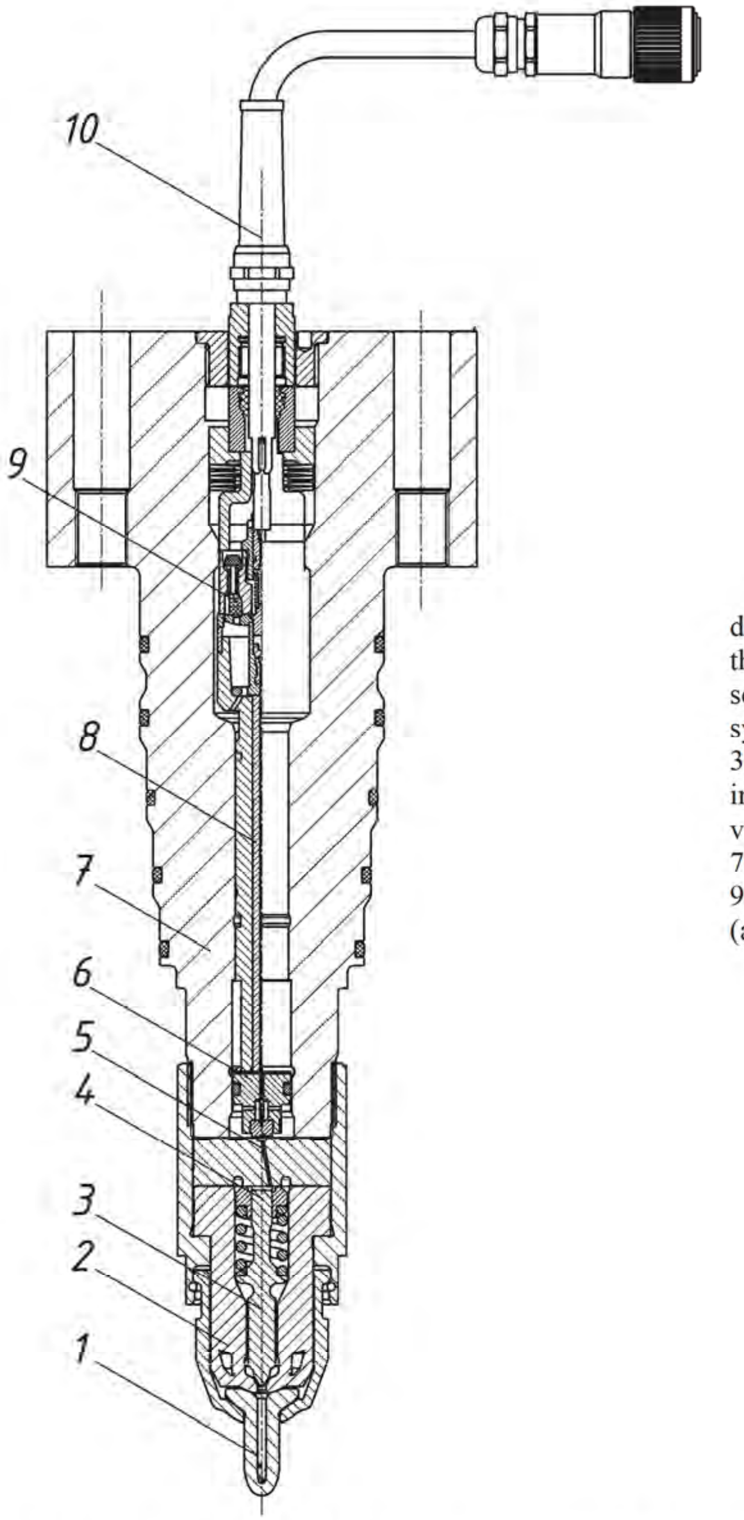


Figure 5.86 – An injector with electrohydraulic control used in low-speed engines of the X-35 series from WinGD equipped with a second generation accumulator fuel injection system: 1 – nozzle tip; 2 – nozzle housing; 3 – nozzle shut-off valve; 4 – hydraulic locking piston; 5 – throttle channel; 6 – unloading valve of the hydraulic shut-off chamber; 7 – fuel injector housing; 8 – pusher rod; 9 – solenoid valve; 10 – electrical connector (adapted from [57])

Power is supplied to the injector solenoid coil through the amplifier module (Common Rail Injection System Driver – CRISD). The CRISD module is powered and controlled by the universal controller (Multi-Purpose Controller – MPC). When an injection command is given, a relatively large current is required to raise the control valve actuator armature. After opening, to keep the valve open, the current density supplied to the solenoid is reduced. The supply voltage during opening is 100V DC, and the power control is carried out using pulse width modulation method.

5.7.3 Accumulator injection systems for medium-speed diesel engines

In recent years, leading manufacturers of marine medium-speed diesel engines have undertaken significant research and development work to develop accumulator injection systems for medium-speed engines. As a result of this work, a fundamentally new class of diesel engines has appeared

tion valve designed to fill the space above the plunger with fuel.

To control the performance of the pumps, regulation is applied by throttling the fuel at the inlet. This method is not highly accurate, but in this case it is not necessary, since the cyclic portion is measured not by the pump itself, but by the control valve, which receives signals from the electronic controller. The microprocessor controller also controls the filling throttle valves, which have an electromagnetic drive and are mounted directly on the pump housing (Fig. 5.89).

In addition to the suction valve, a delivery valve is installed in the pump cover, separating the above-plunger cavity and the high-pressure line in the absence of discharge.

To supply the engine with fuel under high pressure, several pump sections are installed, which are driven from a cam shaft with three-cam washers installed on it (Fig. 5.90).

The cam shaft is driven through a gear transmission from the engine crankshaft. The number of pumps installed on one engine depends on the number of its working cylinders.

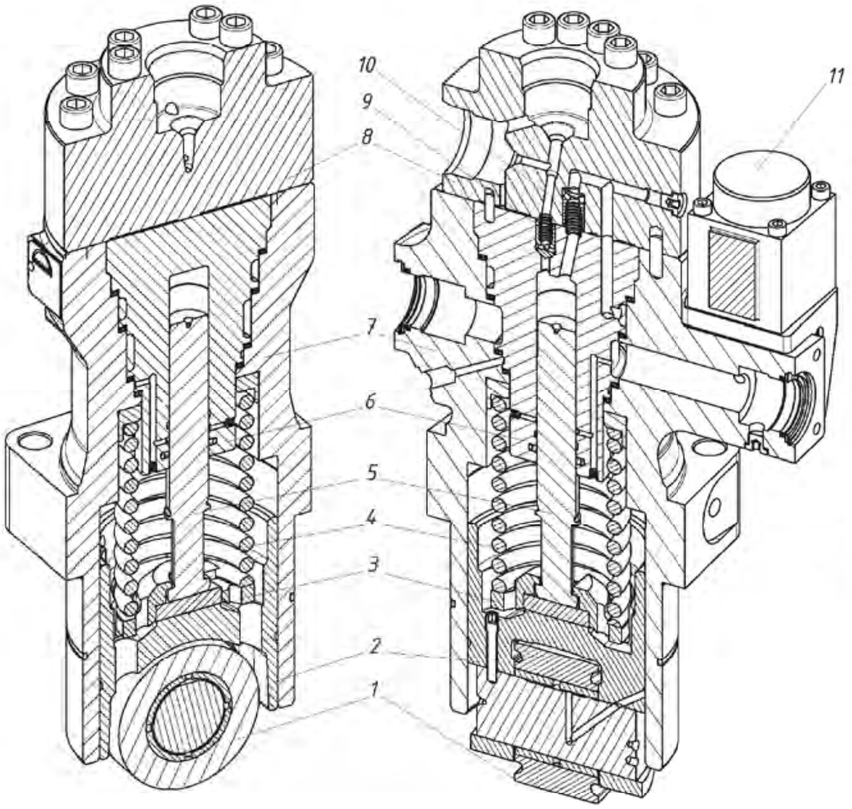


Figure 5.89 – High-pressure fuel pump for MAN series 32/44CR engines: 1 – pusher roller; 2 – pusher; 3 – lower plate of the return spring; 4 – return spring; 5 – plunger; 6 – plunger bushing; 7 – housing; 8 – cover; 9 – delivery valve; 10 – suction valve; 11 – flow control throttle valve with electromagnetic drive (adapted from [60])

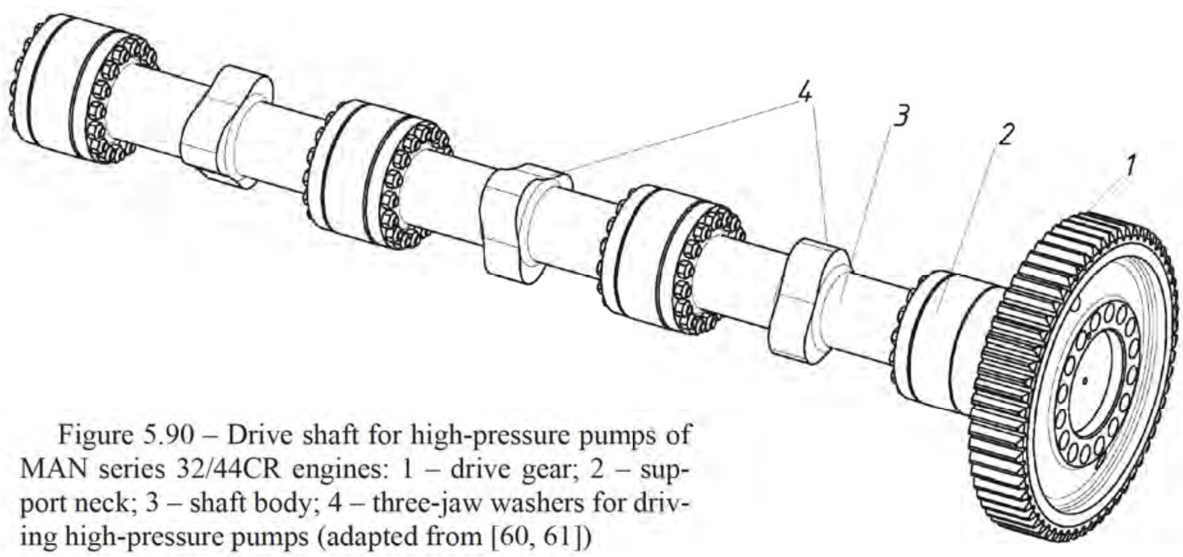


Figure 5.90 – Drive shaft for high-pressure pumps of MAN series 32/44CR engines: 1 – drive gear; 2 – support neck; 3 – shaft body; 4 – three-jaw washers for driving high-pressure pumps (adapted from [60, 61])

formance of the system as a whole.

In addition, the valve prevents the occurrence of wave processes in the pressure line. When the pressure wave reaches the valve, it opens and transfers part of the fuel into the damper cavity, in a large volume of which the wave is dissipated.

A ball valve loaded with a spring is used as a shut-off element. To adjust the pressure in the drain cavity, adjusting washers are placed under the spring.

The bleeding and emergency pressure relief valve is made in one block with the maximum pressure limiting valve (Fig. 5.96).

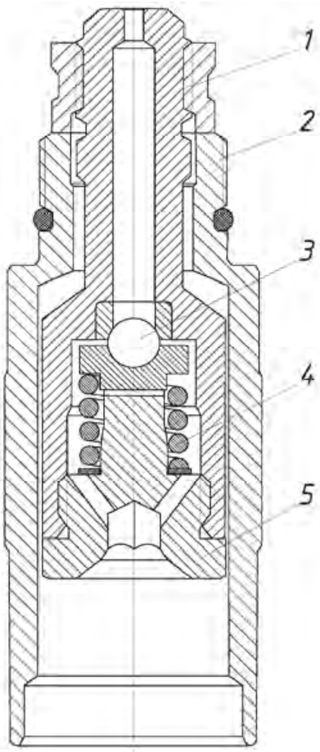


Figure 5.95 – Check valve of the fuel drain system: 1 – valve housing; 2 – outer housing; 3 – ball valve; 4 – pressure spring; 5 – screw plug (adapted from [60])

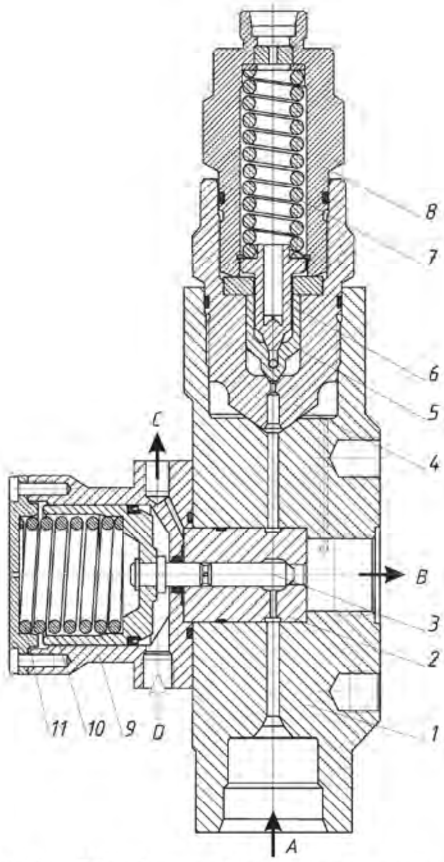


Figure 5.96 – Bleeding and emergency pressure relief valve and maximum pressure limitation valve: 1 – valve block housing; 2 – bleeding valve housing; 3 – bleed valve stem with shut-off cone; 4 – housing of the maximum pressure limiting valve; 5 – plug-in valve-limiter seat; 6 – shut-off cone of the limiting valve; 7 – loading spring; 8 – threaded plug; 9 – pneumatic cylinder for pumping valve drive; 10 – piston of the pumping valve drive; 11 – loading spring; *A* – fuel supply from the accumulator; *B* – drain line; *C* – collection of leaks; *D* – compressed air from the control system (adapted from [60, 61])

The valve is driven by a pneumatic piston, to which air is supplied from the control system.

The valve block is installed in the system at the outlet of the accumulator furthest from the pumps.

When the valve opens, the high-pressure cavity is connected to the drain line and the pressure in the system is released.

A valve is used to pump the fuel system with heated fuel in preparation for starting the engine on heavy fuel.

Opening the valve causes the pressure in the system to become lower than in the supply line. Under the influence of a pressure difference, the heated fuel is divided into two streams. The first flow through the open throttle control valves of the pumps, through the suction valves fills the fuel

To prevent the plunger from being subjected to lateral forces that occur when the pusher roller rolls onto the cam washer, the pump has a separation between the drive mechanism and the pump section. Thanks to this, only forces directed along its axis are transmitted to the plunger.

The pump supply is controlled using throttling valves with an electromagnetic drive (Fig. 5.100 *c*). The flow area at the pump inlet is changed by moving the control cone along the vertical axis. The throttling valve is controlled by an electronic unit, which, depending on the operating mode, changes the flow of the pumps and the pressure under which the fuel enters the system.

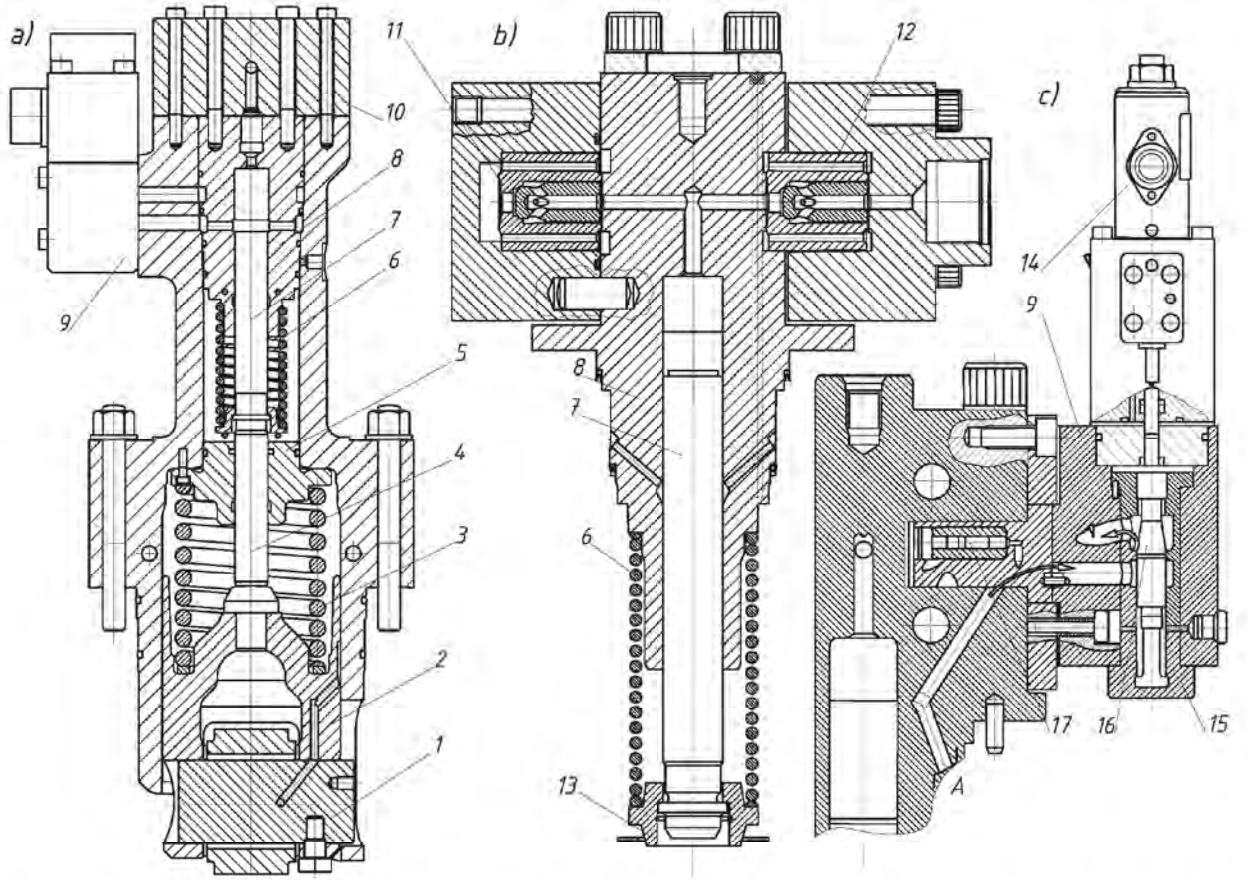


Figure 5.100 – High pressure fuel pump of the 32 series engine from Wärtsilä (*a*), pump section of the fuel pump (*b*) and throttle valve for controlling the fuel supply of 46 series engines from Wärtsilä (*c*): 1 – roller pusher axis; 2 – pusher; 3 – pusher return spring; 4 – pusher rod; 5 – dividing diaphragm with pusher rod guide; 6 – plunger return spring; 7 – plunger; 8 – plunger bushing; 9 – flow control throttle valve; 10 – pump cover; 11 – suction valve; 12 – delivery valve; 13 – lower plate of the plunger return spring; 14 – throttle valve drive solenoid; 15 – throttle valve housing; 16 – throttling cone of the supply control valve; 17 – safety valve; *A* – low pressure fuel supply (adapted from [64, 65])

The pressure accumulator is a thick-walled pipe made of high-quality steel, closed on both sides with covers that are attached to the housing with bolts (Fig. 5.101 *a*). All connecting fittings and various types of valves are concentrated on the accumulator covers. This makes it possible to increase the strength of the housing and ensure its reliable operation at sufficiently high pressures.

To maintain a given operating pressure in the fuel supply system, a pressure limiter regulator is installed on a number of batteries (Fig. 5.101 *b*).

On six-cylinder engines, one regulator is installed; on engines with a large number of cylinders, one regulator is installed per 5-6 cylinders.

Pressure limiter regulator (Fig. 5.101 *b*) is a multifunctional valve system that performs the following functions:

- together with the flow control throttle valve, maintains the specified pressure in the fuel system;
- protects the accumulator and fuel system from excessive pressure increases;

The fuel injector includes three main valves:
- starting, with electromagnetic drive;
- main, hydraulically driven from the starting valve;
- needle valve of the nozzle.

In addition, the fuel injector design includes a valve that limits the residual fuel pressure in the working cavity.

Wärtsilä has used a two-level control system in its accumulator system.

At the first level there is a start valve, which is controlled by the electronic engine control unit. This valve affects the flow of control oil supplied to it from the control line.

At the second level there is a main valve, which is hydraulically actuated by the control oil system, which is controlled by the start valve.

The design of the main valve allows it to sequentially control the hydraulic locking system of the nozzle needle and the process of fuel injection into the combustion chamber. This prevents premature fuel injection until the pressure in the fuel injector cavity reaches its maximum value. This solution has significantly improved the quality of fuel atomization, especially at the initial stage of the injection process.

The supply control start valve is electromagnetically driven by the electronic engine control unit. The working element is a cone-type locking device, held closed by a loading spring. A general view of the starting and main valve is shown in Fig. 5.106.

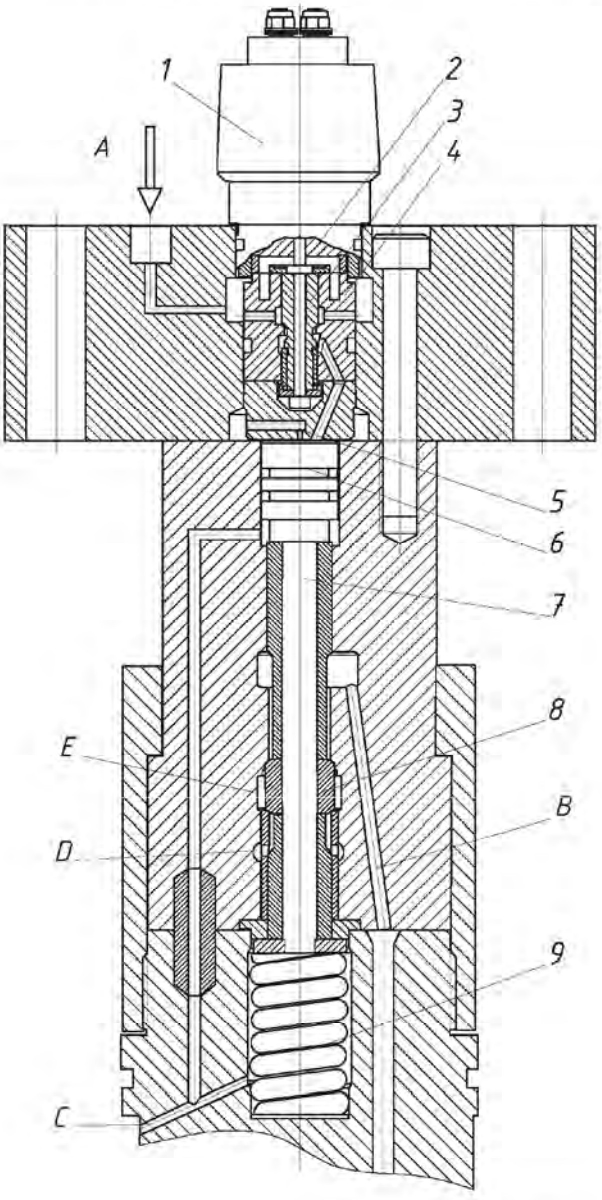


Figure 5.106 – General view of the starting and main valve: 1 – solenoid; 2 – solenoid armature; 3 – start valve; 4 – starting valve housing; 5 – throttle hole; 6 – main valve drive piston; 7 – main valve guide; 8 – main valve; 9 – return spring. *A* – control oil; *B* – fuel supply; *C* – collection of leaks; *D* – fuel outlet for hydraulic locking of the needle; *E* – fuel outlet to the nozzle (adapted from [65, 66])

ondly, by torsional vibrations in the «crankshaft – shaft line» system. In different modes, signals from the sensors may not correspond to the actual position of the shaft. In conditions of such instability, it is difficult to link any control action to the real process. For example, a decrease in the time between passing two nearby control points on the flywheel can be perceived by the control system as the beginning of engine acceleration. The system should respond to this by reducing the fuel supply to the cylinder. But if the decrease in time is caused not by acceleration of the shaft, but by its twisting, a decrease in supply will lead to a decrease in rotation speed, to which the system must respond by increasing supply. As a result, the so-called «engine swing» may occur, in which its speed begins to change cyclically with increasing amplitude. If no measures are taken, this could result in a serious accident.

In the WECS control system, all processes are synchronized relative to a virtual flywheel, which, depending on the operating mode, «rotates» at a constant speed or with constant acceleration. The virtual one is synchronized with the real flywheel only through position and phase sensors. Moreover, synchronization is maintained even if three out of four sensors fail.

Integrated control of the engine's operating process is carried out by changing the supply law in two main positions:

- regulation of the beginning and duration of fuel supply to the working cylinder;
- change in fuel pressure in the accumulator and in front of the nozzle.

The algorithm for changing the above parameters, implemented in Wärtsilä engines, is shown in Fig. 5.111.

Changing the fuel supply advance angle makes it possible to optimize the heat release process relative to TDC, ensuring constancy of the maximum cycle pressure in the load range of 75...100% of the nominal value.

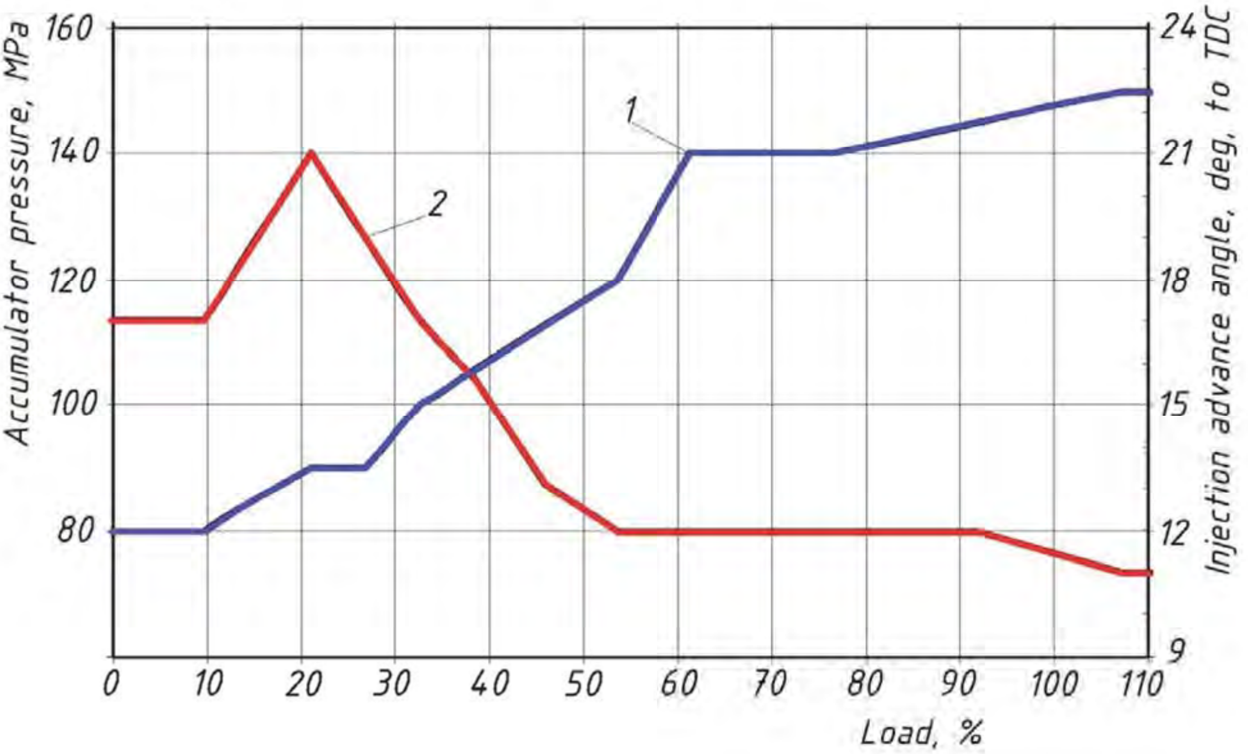


Figure 5.111 – Change in the supply advance angle and pressure in the accumulator as a function of engine load: 1 – pressure in the accumulator; 2 – injection advance angle (adapted from [68])

Regulating the pressure in the accumulator allows you to control the quality of fuel atomization. At low cyclic supplys, a decrease in injection pressure makes it possible to increase the depth of penetration of fuel jets, which improves the distribution of fuel throughout the volume of the combustion chamber. An increase in load leads to the fact that cyclic supplys increase. At constant pressure, this can lead to the settling of a significant part of the fuel on the walls of the combustion

shown in Fig. 5.116, the general arrangement of individual elements on the engine is shown in Fig. 5.117.

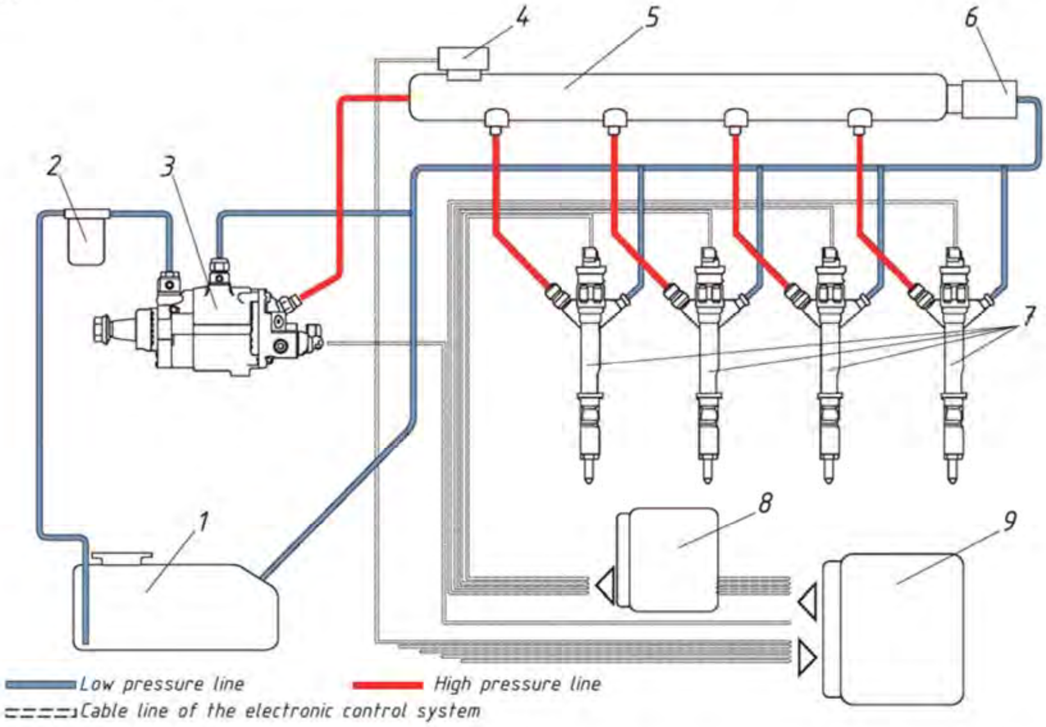


Figure 5.116 – General diagram of the accumulator fuel injection system of a high-speed diesel engine: 1 – fuel tank; 2 – fuel filter; 3 – high pressure fuel pump; 4 – fuel pressure sensor in the accumulator; 5 – accumulator; 6 – fuel pressure regulator in the accumulator; 7 – injectors with electromagnetic control; 8 – injector control unit; 9 – engine control unit (adapted from [71])

To fuel filters have requirements similar to those of traditional fuel systems.

If the engine is supposed to use not only distillate fuels, the system is equipped with fuel heaters and viscosity regulators.

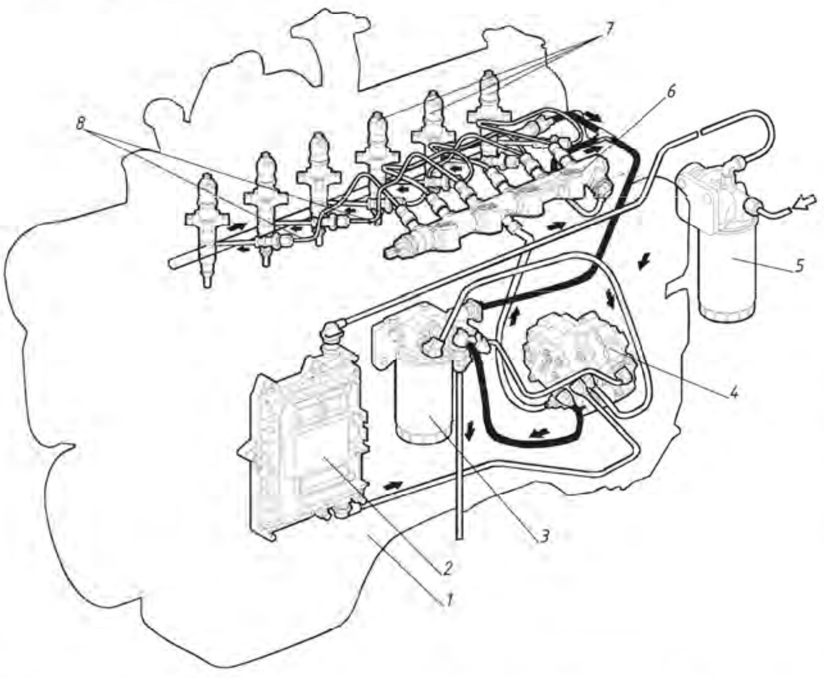


Figure 5.117 – General arrangement of elements of the accumulator fuel injection system of a high-speed diesel engine: 1 – engine; 2 – electronic engine control unit; 3 – fine fuel filter; 4 – pump module; 5 – fuel coarse filter; 6 – accumulator; 7 – injectors with electromagnetic control; 8 – high pressure pipelines (adapted from [72])

Fuel pump block serves to supply fuel to the accumulator at a given pressure. It includes a low pressure module with booster pump and a high pressure module. Further, for simplicity, we will simply call the fuel pump unit a high-pressure fuel pump.

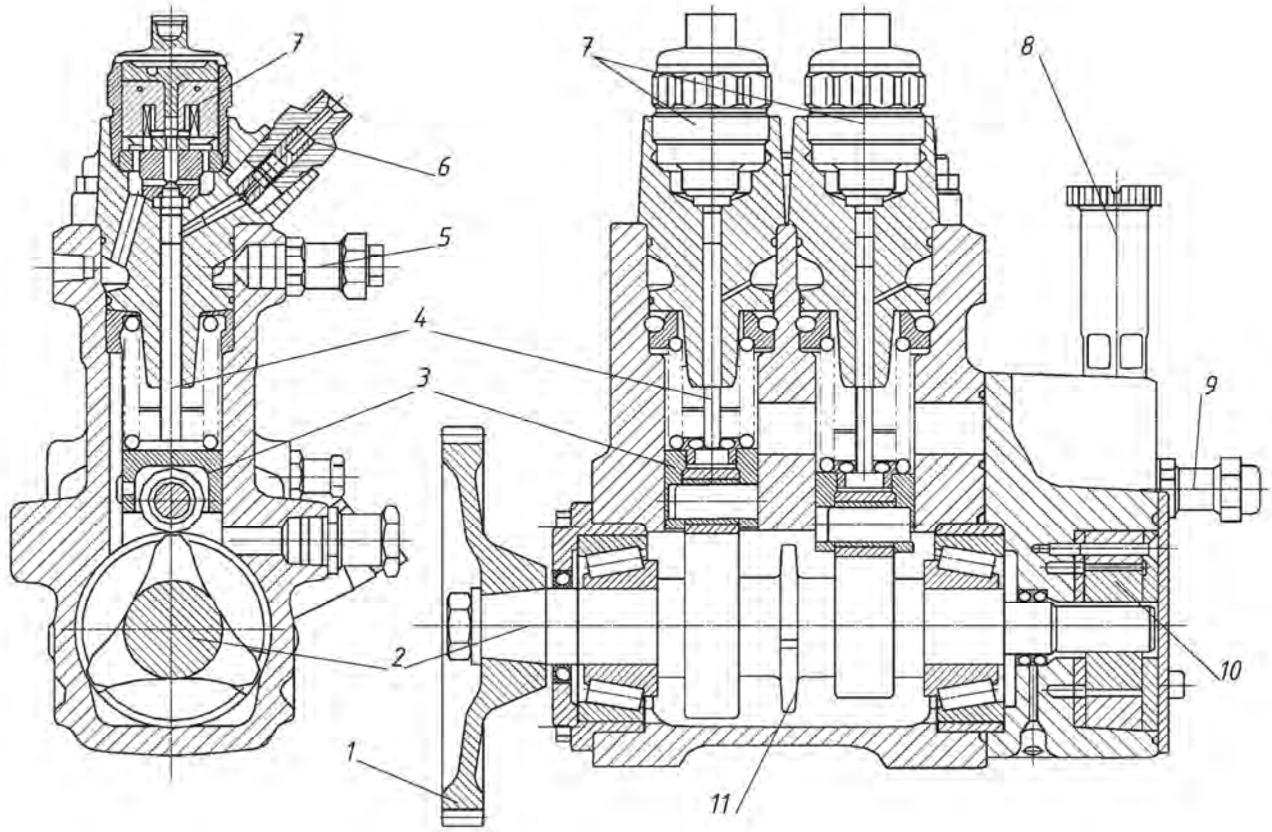


Figure 5.122 – In-line plunger pump type HP0 from Delphi for Common Rail systems of high-speed diesel engines: 1 – drive gear; 2 – drive shaft with cams; 3 – plunger pusher; 4 – plunger; 5 – bypass valve; 6 – high pressure line connection fitting; 7 – solenoid valve for supply control; 8 – booster pump; 9 – pressure limiting valve in the pumping line; 10 – booster pump; 11 – cam with a mark for the speed sensor (adapted from [70])

The supply is controlled using an electrically controlled valve according to the diagram shown in Fig. 5.114, by changing the active stroke of the plunger.

This pump is designed for power supply systems for engines with a number of cylinders from four to eight and with an operating pressure of up to 120 MPa. The pump rotation speed is two times lower than the engine speed. The required supply is ensured by installing cams with different numbers of projections (Table 5.7). Thus, the number of pump strokes coincides with the number of engine operating cycles per revolution of the crankshaft, and the moment of the injection stroke of the plunger is synchronized with the moment of fuel injection. This ensures uniformity of fuel supply and maintenance of the specified pressure in the fuel accumulator.

Table 5.7 – Characteristics of the drive sections of the Delphi type HP0 pump

Number of cylinders	Number of lugs on cam	Number of pump strokes
4	2	4
6	3	6
8	4	8

To reduce friction costs, roller pushers are used to drive the plungers.

Rotary pumps with radial plungers are carried out according to the scheme presented in Fig. 5.123.

The stator, coaxial with the drive shaft, has radial drillings into which precision-fitted plungers with roller pushers moving towards each other are inserted.

During the pumping process, translational motion is imparted to the plungers due to the pusher rollers running onto the radial protrusions of the rotor connected to the pump drive shaft. The return of the plungers to their original position is carried out by springs installed under the pusher.

plunger returns back under the action of a spring. For one revolution of the shaft, three pump strokes are carried out.

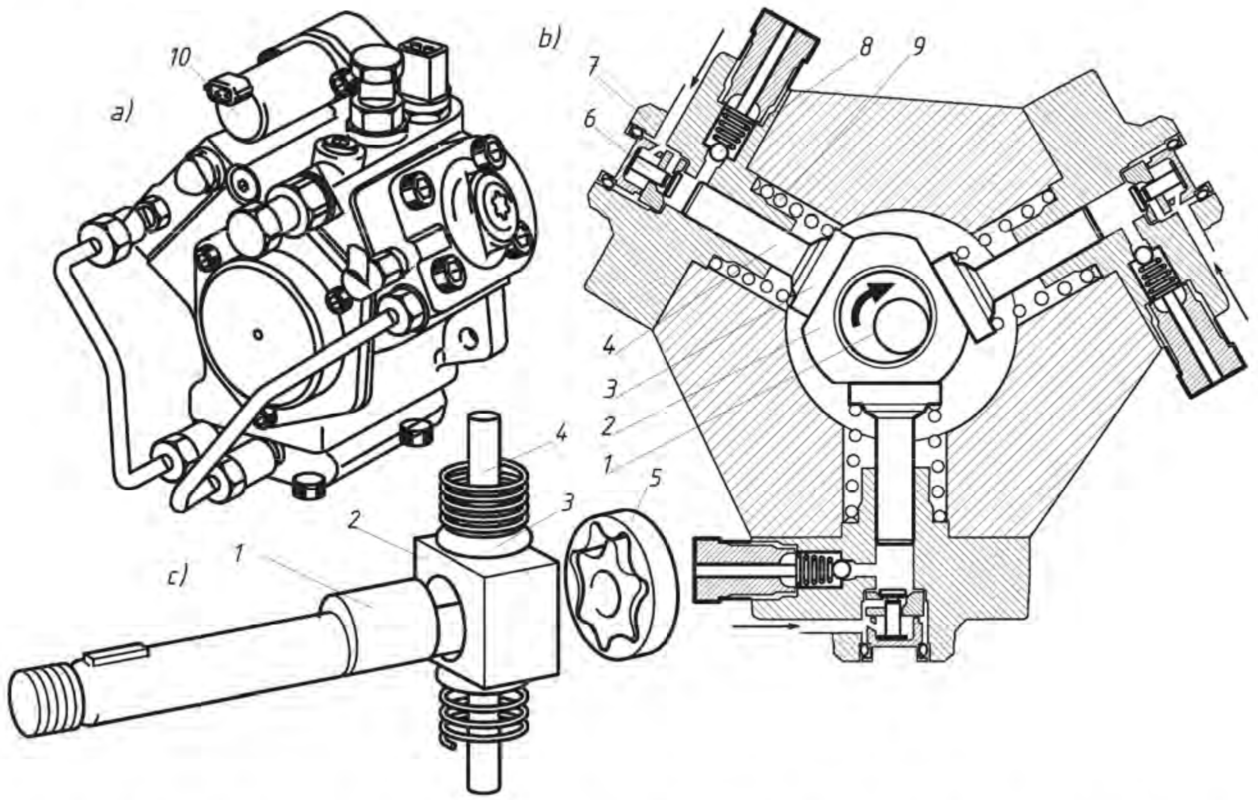


Figure 5.126 – Delphi type DFP3 plunger pump for Common Rail systems of high-speed diesel engines: *a* – general view of a three-plunger pump with pump sections located in the same plane at an angle of 120°; *b* – diagram of a three-plunger pump; *c* – diagram of a pump with an opposed arrangement of pumping sections; 1 – shaft with drive eccentric; 2 – rotor; 3 – plunger pusher; 4 – plunger; 5 – booster pump; 6 – suction valve; 7 – plunger sleeve with cover; 8 – delivery valve; 9 – return spring; 10 – flow control valve (adapted from [74])

The disadvantage of such a drive is the presence of slippage between the rotor flat and the pusher, as a result of which forces arise that distort the plunger in the sleeve and increase the power consumption of the pump drive.

When the plunger moves from BDC to TDC, the suction valve connecting the space above the plunger with the inlet cavity is closed. An increase in pressure in the above-plunger cavity leads to the opening of the delivery valve, and fuel under pressure enters the accumulator. After the plunger passes TDC, the pressure in the space above the plunger drops below the pressure in the accumulator, and the delivery valve closes. Next, when the pressure drops below the pressure in the pumping cavity, the suction valve opens and fuel begins to flow into the supra-plunger cavity until the plunger lowers to BDC. Then the process is repeated.

In a number of pumps designed for the large flow required for stable operation of the engine at full load, at low loads a large amount of fuel is discharged from the accumulator into the return line. In addition, fuel, like any liquid, heats up during compression, and prolonged operation at low loads leads to an increase in its temperature, which leads to a decrease in engine efficiency. To prevent heating and to reduce the cost of pump drive, a number of designs provide for the shutdown of one plunger section. To do this, an electromagnetic valve is installed on it, which, upon command from the electronic unit, presses the inlet valve, keeping it open. Fuel compression does not occur in this section, and pump performance decreases.

If it is necessary to obtain high fuel pump performance, it can be achieved by installing several star-shaped modules. In Fig. 5.128 shows the design of a fuel pump for high-speed high-power en-

The closing part of the valve consists of a ball, which separates the high and low pressure lines, and an armature, which, under the action of a spring, presses it to the seat. The force of the spring loading the valve is such that, under the influence of the set pressure in the accumulator, the valve separates the high and low pressure cavities. If the pressure, for example, due to wave processes in the accumulator, increases above the established norm, the valve overcomes the spring force and bleeds part of the fuel into the drain line. If it is necessary to reduce the pressure in the accumulator, voltage is applied to the solenoid, under the influence of which a current appears in the coil, creating a magnetic field that attracts the armature that holds the ball valve. Thus, this valve functions as both a limiter valve and a pressure regulator valve in the accumulator. It is assigned the role of a second control channel together with pump flow control. The pressure regulator sets the pressure in the accumulator depending on the engine load, rotation speed and thermal state of the diesel engine.

The spring stiffness is selected to limit fluctuations within 10 MPa, and the main adjustment range (23...160 MPa) is carried out using an electromagnet. The force developed by the electromagnet depends on the duration of the current pulse applied to it (pulse width modulation method). The current supplied to the electromagnet winding is constant.

It should be noted that the control method by draining fuel from a high-pressure accumulator has a number of significant disadvantages: it is too uneconomical, requires a special fuel cooler and sufficient speed of the valve itself.

For this reason, if it is possible to circumvent these shortcomings, then the non-optimality of the injection pressure, from the point of view of the requirements for organizing the operating process, is neglected in a number of designs. In such cases, the accumulator is equipped only with a maximum pressure limiting valve, an example of the design of which is shown in Fig. 5.131.

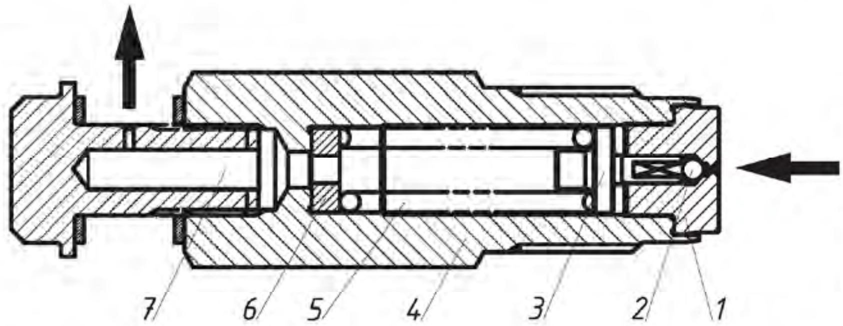


Figure 5.131 – Maximum pressure limiting valve in the accumulator: 1 – ball valve seat; 2 – ball valve; 3 – ball valve pusher; 4 – valve housing; 5 – loading spring; 6 – adjusting washer; 7 – drain line fitting (adapted from [70])

A loading spring presses the ball valve against the seat through a pusher. If the pressure in the accumulator is too high, the valve opens, overcoming the force of the spring, and part of the fuel from the accumulator is discharged through the drain line back into the supply tank.

The valve opening pressure is regulated by changing the tightening of the loading spring by selecting an adjusting washer of the appropriate thickness.

The maximum cyclic flow limiter serves to disconnect the high pressure line from the accumulator, in which depressurization occurs, for example, due to a stuck fuel injector needle or flow control valve. For high-speed engines, the flow limiter valves are usually mounted in the housing of the fitting through which the accumulator is connected to the high pressure line. The operation of the limiter is based on the principle of a pressure difference on both sides of the separating piston. In Fig. 5.132 shows the design of a ball-piston type limit valve and its operating procedure.

As fuel passes through the throttle hole in the separating piston, the cavity of the restrictor valve is filled. The start of fuel supply by the injector leads to a pressure difference between the cavity of the high pressure line and the accumulator. Since the cross-section of the throttle hole cannot compensate for this difference, the separating piston moves to the right (Fig. 5.132) and, compressing the return spring, moves the ball valve. The movement of the piston is proportional to the cyclic fuel supply, and if it does not exceed the maximum permissible value, the ball valve does not reach its extreme position, and after stopping the supply, the valve together with the piston returns to its original position. If the amount of fuel delivered by the injector exceeds the theoretical maximum

not depend on the engine speed (curve 6 in Fig. 5.135).

When the control solenoid valve is closed, all elements return to their original state (Fig. 5.134 *c*).

When the engine is not running, and there is no pressure in the accumulator, the spring presses the fuel injector needle to the seat, closing the fuel injector.

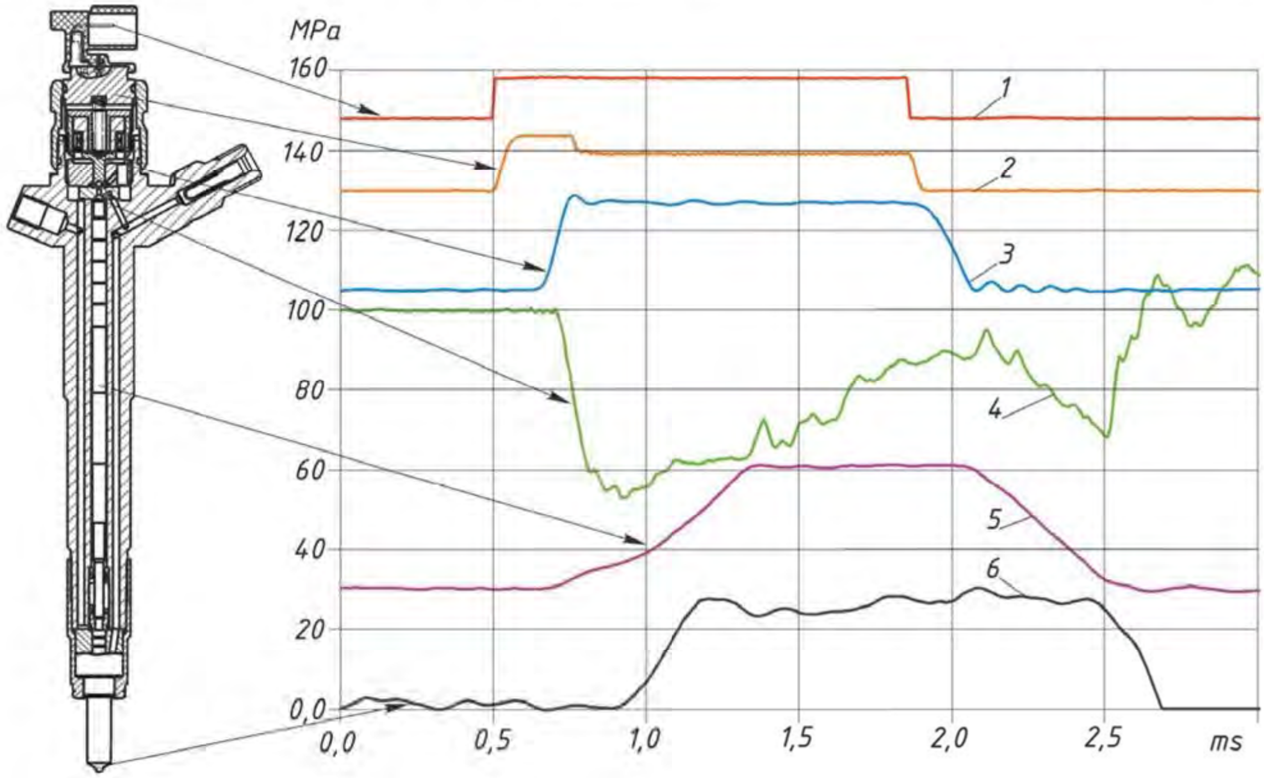


Figure 5.135 – The nature of the main processes during fuel injection using an electrohydraulic injector with throttle control: 1 – signal from the electronic control unit; 2 – solenoid force; 3 – movement of the solenoid armature; 4 – pressure in the needle hydraulic locking multiplier chamber; 5 – movement of the needle valve; 6 – fuel consumption through the fuel injector (adapted from [80])

Indirect needle control is used because the direct force of the solenoid valve is not enough to quickly raise the nozzle needle.

The disadvantage of this method of controlling fuel injection is the presence of the so-called control supply, which is an additional amount of fuel required to control the operation of the injector. After injection is completed, the fuel that controls the supply is discharged into the return line. In the process of opening the fuel injector, the multiplier piston, with its upper end, tends to block the axial throttle hole leading to the ball valve. As a consequence of this, the unloading of the multiplier chamber is reduced, but its filling continues through the fuel supply fuel injector from the accumulator. As a result, the pressure in the intensifier chamber begins to rise again, moving the intensifier piston down, and the pressure in the chamber begins to fall again. As a consequence of this, a self-oscillatory process occurs, which continues until the multiplier piston takes an equilibrium position at some distance from the stop. In this case, the minimum required fuel consumption for control is set (curve 6 in Fig. 5.135). This establishes a feedback relationship between needle lift and control fuel consumption. As a result, the response time of the needle valve and the fuel consumption for control are reduced.

In addition to the control supply, there are fuel leaks through the nozzle needle and the multiplier piston guide. All this fuel is diverted into the return line and returned to the supply tank. Other components of the injection system are also connected to the return line.

The presence of fuel consumption for control leads to the need for additional power consumption to drive high-pressure pumps. Compared to traditional fuel supply systems, this increase can be up

its drive uses a mechanical lever displacement multiplier (Fig. 5.138 *b*, *c*).

When voltage is applied, the pressure plate of the piezoelectric element unit activates the multiplier. At the beginning of its stroke, the maximum force is transmitted through it, counteracting the high pressure of the valve opening, while the gear ratio of the drive is almost equal to unity ($a/b \approx 1$, Fig. 5.138 *b*). At the end of the stroke, the force decreases and the stroke increases ($a/b > 1$, Fig. 5.137 *c*), which ensures the valve moves by almost 0.1 mm.

A promising direction for improving accumulator fuel supply systems is the use of injectors with direct drive of a needle valve from piezoelectric elements. This solution allows you to maximize the speed of the injector, since the response speed of the piezo actuator is several orders of magnitude higher than the response speed of electromagnetic systems. In addition, the piezo actuator is capable of developing significant forces sufficient to keep the nozzle valve closed.

The main obstacle to the implementation of direct piezo drive technologies remains the relatively small movement of piezo elements, which is insufficient to ensure efficient injection of large cyclic supplys. Therefore, the scope of use of these technologies in the near future will be limited primarily to small-sized engines for ground vehicles.

As in the previously discussed MSE and HSE accumulator systems, the fuel supply for injectors with a piezo drive is controlled by an electronic control system that sets the start point and duration of injection. In this type of injectors, it is possible to vary the fuel supply law over a wider range by changing the voltage supplied to the piezo drive plates.

5.8 Fuel equipment of gas and gas-diesel marine engines

The specific operating conditions of ships have left their mark on the development of fuel systems for marine engines running on gas. This is primarily due to the need to maintain the ability to operate the engine on liquid fuels, which arises whenever the ship is moving in ballast. In addition, depending on the type of cargo, sailing conditions and time, the composition of the gases used in the power plant can change significantly, and therefore the fuel system must adequately respond to such changes and ensure the operation of the engines at nominal modes. Based on this, the bulk of marine engines today are created as dual-fuel (DF) engines, that is, capable of running on gas, liquid fuel, or both fuels at once in different proportions.

There are several fundamentally different approaches to organizing the operating process in gas engines [83]:

- converting diesel engines into engines with external mixture formation and spark ignition, operating according to the Otto cycle;
- use of external mixture formation with ignition of the gas-air mixture from a small portion of liquid fuel injected into the working cylinder;
- use of internal mixture formation and ignition of the gas-air mixture from a small portion of liquid fuel injected into the working cylinder.

Each of the above methods can be implemented using both natural and petroleum gases. All these methods have found their application in various types of marine engines, so we will consider the features of the fuel systems of each type of engine in more detail. Issues related to the storage of gases on board a ship will not be considered, since they are beyond the format of this book. This topic is discussed in sufficient detail in the relevant specialized literature.

5.8.1 Converting diesel engines to engines with external mixture formation and spark ignition

Converting diesel engines into engines with external mixture formation and spark ignition is the easiest way to convert an engine to gas fuel. Its advantages include:

- simplification of the design (if there is no need to maintain dual fuel, you can completely abandon the liquid fuel injection system, replacing it with simpler ones: an air-gas mixing system and a spark ignition system);
- operation of the supply system at low pressures, which reduces the safety requirements for the

second stage of pressure reduction when using compressed natural gas. The pressure at the regulator inlet is within 2420 kPa, at the outlet – 21 kPa.

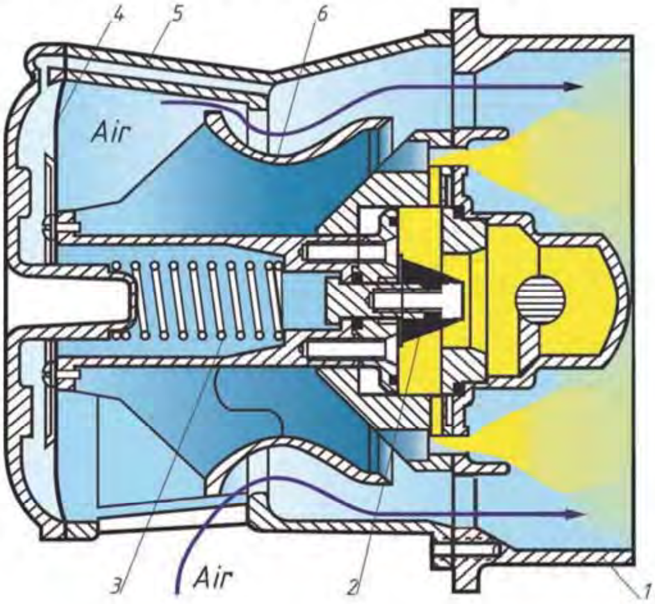


Figure 5.143 – Gas mixer for Caterpillar G 3500 series engines: 1 – mixer housing; 2 – gas valve; 3 – gas valve spring; 4 – gas valve drive diaphragm; 5 – channel for supplying vacuum under the diaphragm of the gas valve drive; 6 – diffuser (adapted from [86])

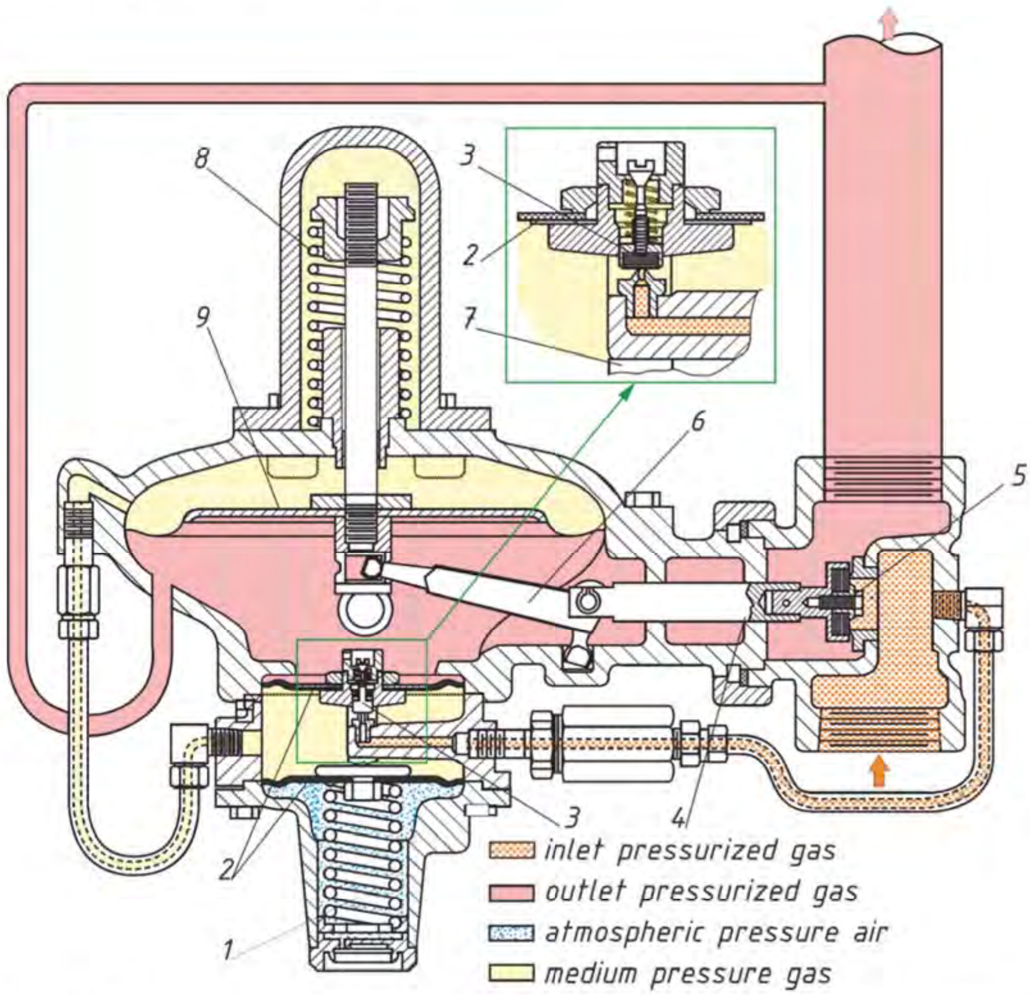


Figure 5.144 – Design and operating diagram of a dual gas reducer for a Caterpillar G 3600 series engine: 1 – operating pressure reducer spring; 2 – working pressure reducer diaphragms; 3 – double valve of the working pressure reducer; 4 – drive rod of the outlet pressure adjustment valve; 5 – outlet pressure adjustment valve; 6 – drive lever for the outlet pressure adjustment valve; 7 – intermediate stop of the operating pressure reducer valve drive; 8 – outlet pressure reducer spring; 9 – output pressure reducer diaphragm (adapted from [86])

system, and the other is equipped with a solenoid valve for controlling the flow, typical for injectors in accumulator systems. The nozzle housing also has two needle valves: a small one for pilot fuel injection and a large one for main fuel injection. The pilot injection is controlled by the electronic engine control unit.

The transition from gas to diesel fuel is also controlled by the control unit. If necessary, you can switch the engine from gas to diesel fuel without stopping it at full load. The reverse transition can be carried out at a load of up to 80% of the nominal load.

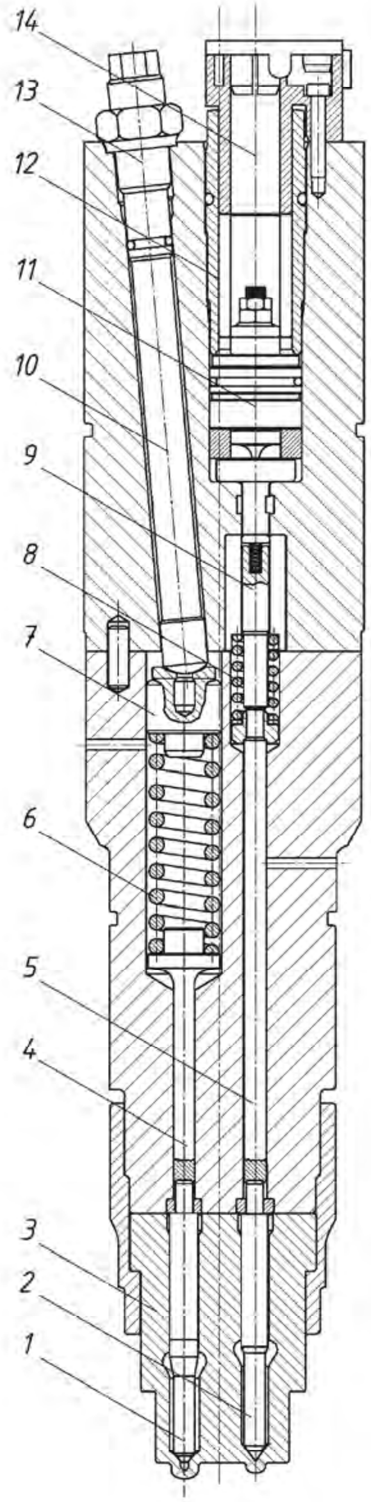


Figure 5.148 – Injector of a gas-diesel engine 50DF from Wärtsilä with a dual fuel injector for reserve and pilot liquid fuel: 1 – needle valve for supplying reserve fuel; 2 – pilot fuel supply needle valve; 3 – nozzle housing; 4 – reserve fuel needle valve rod; 5 – pilot fuel needle valve rod; 6 – main fuel needle valve spring; 7 – spring stop; 8 – spring of the ignition fuel needle valve; 9 – piston for locking the pilot fuel needle valve; 10 – spring adjusting rod for setting the opening pressure of the reserve fuel valve; 11 – pilot fuel nozzle control solenoid; 12 – installation cup; 13 – adjusting bolt; 14 – installation stop with electrical connector (adapted from [89])

The gas supply system is controlled by an electronic unit. On gas-diesel engines for marine use, the gas supply system diagram shown in Fig. 1 is widely used. 5.149.

the symbol GA.

The low-pressure gas supply technology developed for a dual-fuel low-speed engine is designed to burn a lean air-fuel mixture. Previously, this technology was tested by the company on four-stroke medium-speed engines. As shown in Fig. 5.151, gas enters the cylinder after all gas exchange organs are closed, but the pressure still remains relatively low. In practice, gas supply valves are installed at a certain height from the purge ports to provide the necessary time for filling the cylinder with gas until the piston passes the filling holes.

The supply of gas fuel during the compression stroke allows it to be supplied to the engine cylinders at a relatively low pressure of 1.0...1.6 MPa versus 30 MPa when gas is supplied at the end of the compression stroke. This circumstance makes it possible to significantly simplify the fuel system and use cheaper and more reliable screw or centrifugal compressors instead of piston ones used in GI systems. The presence of low pressure in gas lines reduces the risk of gas leaks, and therefore makes them safer.

During the compression process, the gas is well mixed with air and ignited with the help of a pilot portion of fuel. In this case, the size of the pilot fuel portion does not exceed 1% of the cyclic supply of liquid fuel at full load and remains unchanged in all operating modes of the engine on gas fuel The use of natural gas makes it possible to reduce the formation of NO_x by 90% compared to the use of liquid fuel, which is explained by a more uniform temperature distribution throughout the combustion chamber and a smaller number of high-temperature zones in which the formation of NO_x occurs. The use of this technology makes it possible to comply with the International Maritime Organization IMO Tier-III requirements for NO_x emissions without any post-engine exhaust gas treatment.

To ensure dual-fuel operation of the engine, it is equipped with three independent fuel supply systems, each of which can be controlled by an electronic microprocessor module according to a separate program, depending on the fuel used and the engine operating mode.

To supply reserve liquid fuel, a standard accumulator-type fuel system is used, which is typical for all engines of the RT-flex series (clause 5.67). At the same time, the engine retains the ability to operate on heavy fuels over the entire range of load and speed conditions.

The pilot fuel is supplied by a separate low-capacity accumulator system (Fig. 5.153).

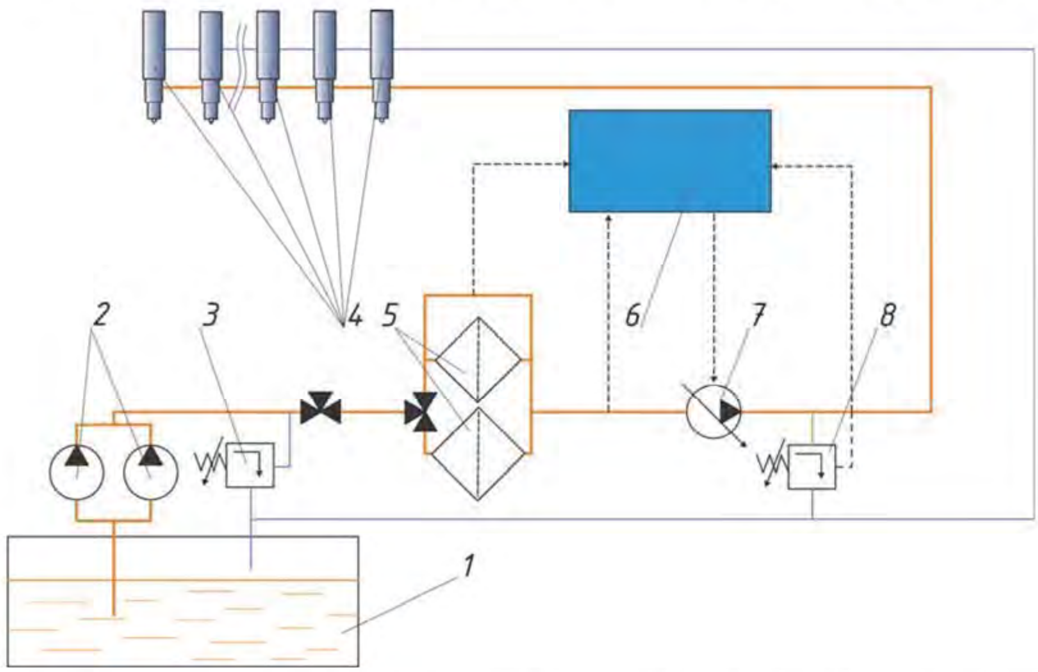


Figure 5.153 – Diagram of the pilot fuel accumulator system of the RT-flex DF gas-diesel engine: 1 – pilot fuel tank; 2 – booster pumps; 3 – bypass valve of the low pressure line; 4 – fuel injectors for pilot fuel injection; 5 – fine filters; 6 – electronic control unit; 7 – high pressure fuel pump; 8 – bypass valve of the high pressure line; —— hydraulic lines; - - - - control lines (adapted from [83])

At the same time: the cost of work for compressing the mixture in which heat is already being released increases; mechanical loads on engine parts increase; vibration leads to accelerated destruction of engine structural elements. To avoid detonation, it is necessary to maintain the composition of the lean gas-air mixture in a fairly narrow range (Fig. 5.139), which entails a decrease in the energy potential of the charge and, consequently, a decrease in the work it produces.

The resistance of gas fuel to detonation, that is, its ability to resist self-ignition during compression, is determined by the methane number.

The rate of physical and chemical processes leading to detonation depends on the composition of the gas-air mixture. To operate the engine at rated power, the methane number of the gas should not be less than 80; when the power is reduced, the methane number can also be reduced to 80...55 units (Fig. 5.159 *a*). In addition, prolonged compression of the gas-air mixture, accompanied by an increase in its temperature, makes the operating process extremely sensitive to the air temperature at the engine inlet (Fig. 5.159 *b*). The higher the temperature at the beginning of the compression process, the faster the physicochemical processes occur, leading to detonation combustion.

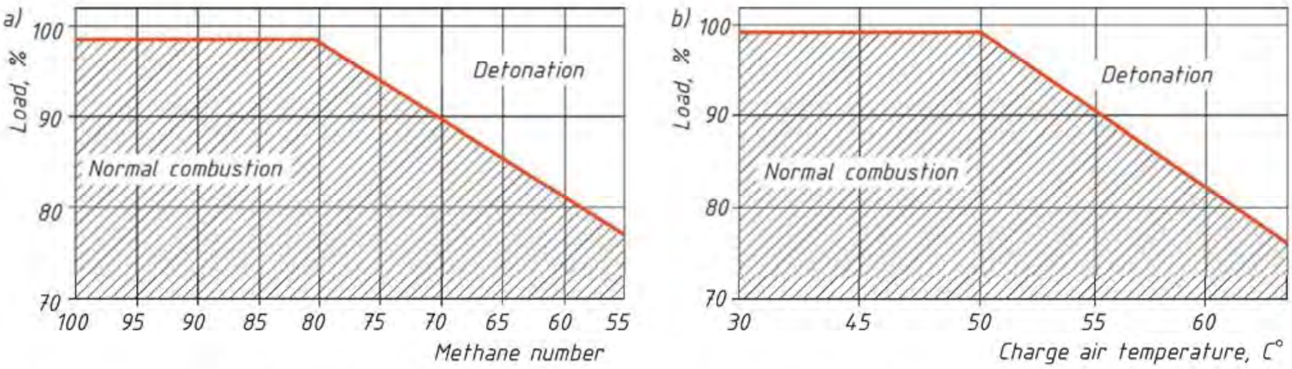


Figure 5.159 – Influence of methane number (*a*) and charge air temperature (*b*) on the occurrence of detonation combustion in the working cylinder of a low-speed engine with gas supply on the compression stroke (adapted from [100])

The need to maintain the composition of the gas-air mixture in a fairly narrow range (Fig. 5.139) does not allow one to quickly increase engine power by significantly increasing the gas supply to the working cylinder, as this can lead to misfires. Reducing the load should also not be accompanied by a sharp decrease in gas supply, as this will lead to detonation combustion. Thus, engines with gas fuel supply on the compression stroke are characterized by a slow response to load changes. The transition from mode to mode should be carried out gradually with delays in intermediate modes to stabilize the composition of the gas-air mixture (Fig. 5.160). This feature makes it difficult to use the engine during maneuvers and requires its conversion to liquid fuel. At the same time, it should be noted that environmental requirements are most stringent in coastal areas, where it is often necessary to maneuver, which to some extent neutralizes the benefits of using gas fuel.

In order to prevent engine operation with detonation, special sensors are installed on each cylinder that record wave processes in the cylinder. If detonation occurs, the sensors transmit a signal to the electronic control unit, which reduces the load on the engine or switches it to liquid fuel. The transition from one type of fuel to another can be carried out without stopping the engine at powers up to 80% of its rated power on liquid fuel. The joint parallel operation of liquid and gaseous fuel supply systems is considered as a promising direction. In this case, part of the energy in the cylinder is obtained by burning gas fuel, which is supplied in the amount necessary to ensure knock-free compression, and the missing part of the energy is obtained by injecting liquid fuel into the combustion chamber.

In addition to knock sensors, the engine is equipped with pressure sensors in the working cylinder, the main task of which is to monitor misfires. If they occur, unburned gas-air mixture may accumulate in the exhaust receiver, which can lead to an explosion and engine damage. Misfires or inefficient fuel combustion can result from malfunctions in the gas-air mixture pilot ignition sys-

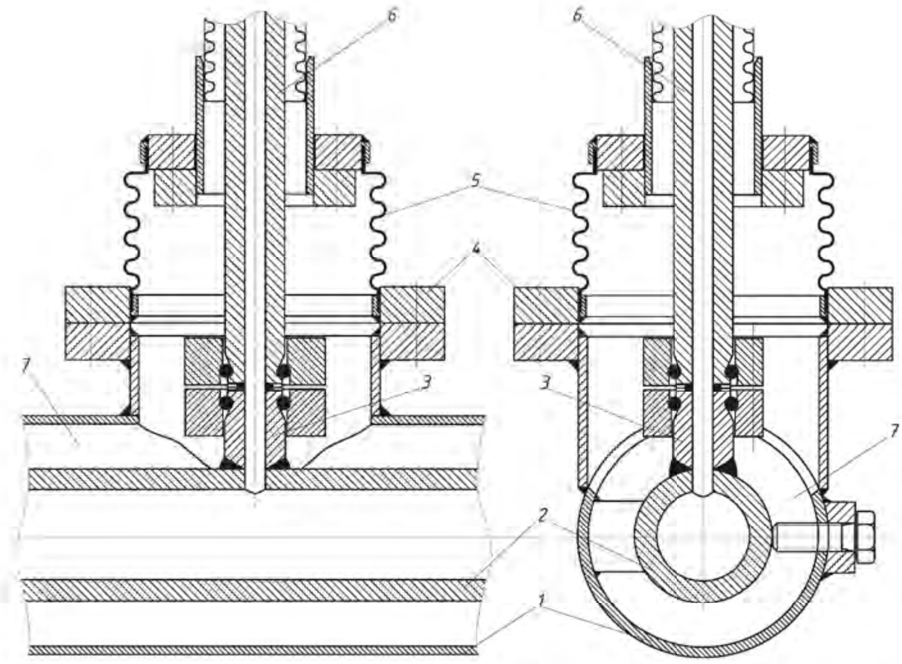


Figure 5.164 – Fragment of the gas line of a gas-diesel engine of the ME-GI series: 1 – protective shell; 2 – gas main; 3 – connecting fitting; 4 – connecting flange; 5 – protective corrugated shell; 6 – gas outlet pipes to the supply control unit; 7 – ventilated space (adapted from [103])

To increase the safety of engine operation, the power plant includes a system of inert gases, which makes it possible to purge both the entire gas fuel supply system and its individual elements under a pressure of 0.4...0.8 MPa. Such cleaning is a mandatory procedure when switching to a diesel cycle or if any part of the gas supply system is damaged.

As already noted, gas-diesel engines are retrofitted with a system for supplying gas fuel to the working cylinder. The power supply system for a gas-diesel engine is shown schematically in Fig. 5.165 *a*.

From Figure 5.165 *b* it is clear that gas is supplied to the combustion chamber immediately after the pilot portion of liquid fuel is supplied to the cylinder and ignited. In this way, a high degree of fuel burnout is achieved and the danger of gas entering the sub-piston space through leaks in the piston rings is prevented.

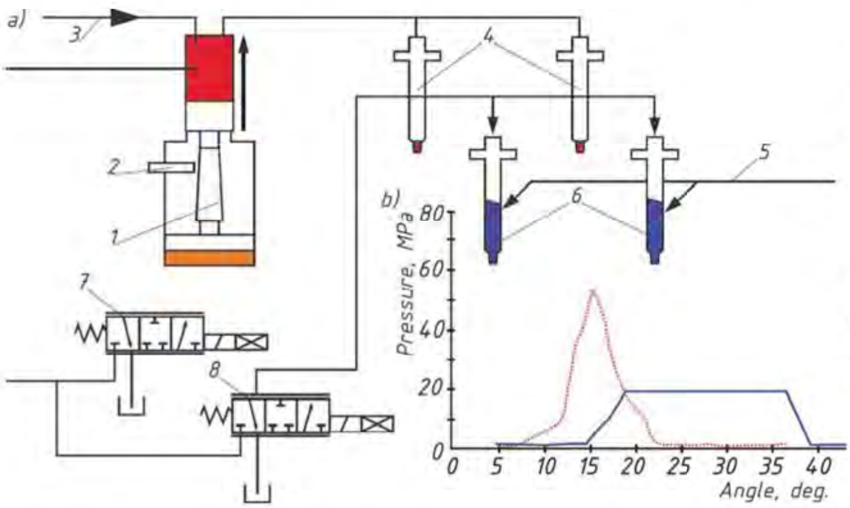


Figure 5.165 – Diagram of the fuel system of a gas-diesel engine of the ME-GI series (*a*) and the laws of supply of pilot (- - - -) and gas fuel (——) (*b*): 1 – injection pump with hydraulic drive; 2 – liquid fuel supply quantity sensor; 3 – liquid fuel supply; 4 – fuel injectors for liquid fuel injection; 5 – gas supply; 6 – gas injectors; 7 – spool valve for fuel injection pump control (FIVA); 8 – spool valve for controlling the hydraulic drive of injectors (ELGI); 9 – supply of control oil under pressure 30 MPa (adapted from [104])

All gas supply control elements are assembled in one module, which includes: a gas accumulator, a main shut-off valve with a hydraulic drive, a valve for purging the system with inert gas, and a valve for controlling the hydraulic drive of the injectors.

The module itself is attached to the cylinder cover, which has internal drillings for supplying gas from the control module to gas injectors installed in the cylinder cover next to the liquid fuel injection fuel injectors.

The general structure of the gas supply control module is shown in Fig. 5.166. The module dia-

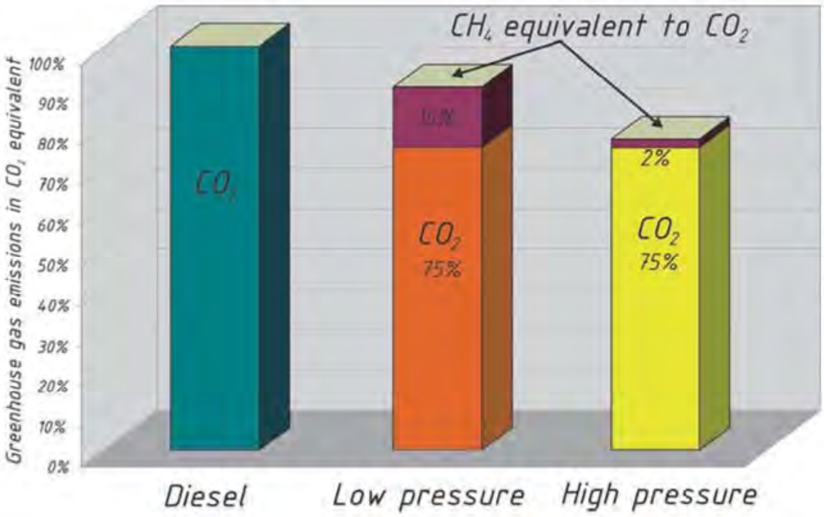


Figure 5.170 – Greenhouse gas emissions (CO_2+CH_4 in CO_2 equivalent) from exhaust gases of gas-diesel engines with low-pressure (LP) and high-pressure (HP) systems in relation to a diesel engine (adapted from [94])

- engine power limitation at 80% of the nominal when operating on gas fuel;
- slow response to load changes;
- higher specific gas fuel consumption compared to high-pressure systems, especially when the load decreases (Fig. 5.171).

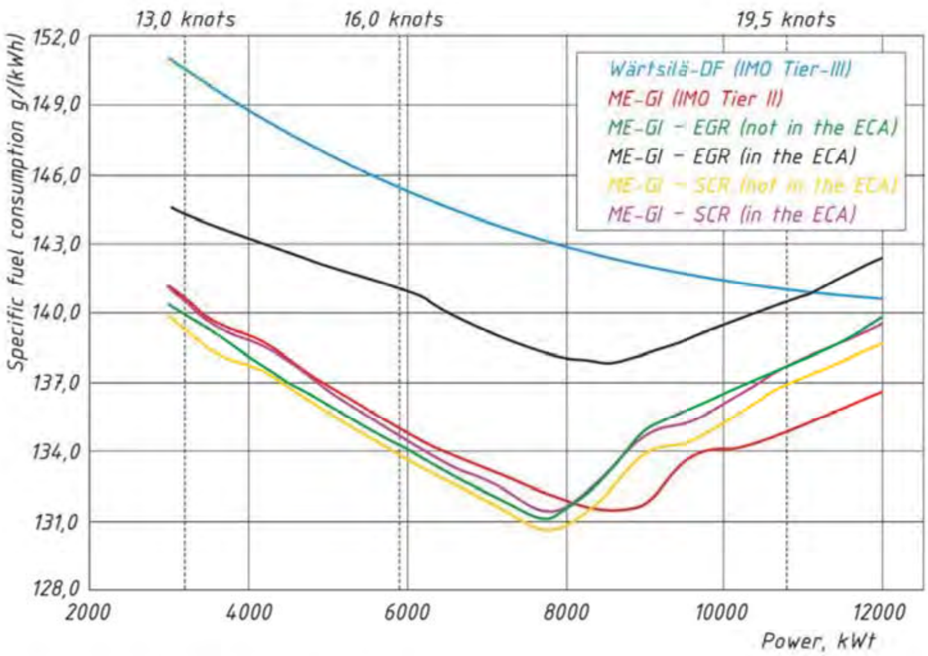


Figure 5.171 – Change in the corrected specific gas fuel consumption for main engines with low (Wärtsilä DF) and high pressure (ME-GI) systems, taking into account the ignition fuel corrected to gas fuel. EGR – Exhaust Gas Recirculation; SCR – Selective Catalytic reduction; ECA – Emission Control Area (adapted from [108])

Advantages of direct injection systems:
- when gas fuel is supplied directly into the combustion chamber, the occurrence of detonation can be completely eliminated, therefore, there are no restrictions on engine power when operating on gas fuel, and the requirements for the quality of gas fuel are less stringent;
- gas fuel cannot enter the sub-piston space;
- lower greenhouse gas emissions compared to low-pressure systems(Fig. 5.170);
- lower specific gas fuel consumption compared to low-pressure systems, especially at partial loads (Fig. 5.171).

Disadvantages of direct injection systems:
- the use of high pressure gas complicates the fuel system and increases the requirements for its safety;
- high consumption of liquid fuel for pilot ignition of the gas-air mixture;
- to compress natural gas, it is necessary to use multi-stage compressors, which reduces the economic performance of the power plant due to the increasing costs of compressing gas fuel to high

5.9 Mixture formation in diesel engines. Performance indicators of the fuel equipment of modern diesel engines and their impact on the operating process

The mechanical and thermal stress of the parts of the cylinder-piston group, the crank mechanism, the distribution of power between the cylinders, and ultimately the efficiency, reliability and service life of the diesel engine largely depend on the quality of the fuel system. The dynamics of heat release in the cylinder directly depends on the perfection of the design of the fuel equipment, which significantly determines the nature of the engine's operating process, its economic and environmental characteristics.

The main factors influencing the process of heat release in the engine are the chosen method of mixture formation, the characteristics of the process of supplying fuel to the combustion chamber and the quality of mixture formation.

Factors influencing the quality of mixture formation and combustion of fuel can be divided into independent and dependent on the performance of fuel equipment. The first group of factors includes: the quality of the engine air supply, the chosen method of mixture formation, the degree of mobility of the air charge and its distribution relative to the jets of injected fuel (determined by the shape of the combustion chamber). The second groups of factors that have the greatest impact on the operating process of a diesel engine include: the quality of fuel atomization, injection start angle, injection characteristics and duration.

Next, we will consider in more detail the second group of factors as being directly related to the topic of this chapter. Issues related to the first group of factors can be studied in specialized literature.

5.9.1 Fuel injection and atomization processes

In the power supply systems of modern diesel engines, liquid fuel supplied to the nozzle under pressure is the main source of energy needed to produce an aerosol.

Spraying is the result of internal disturbances in the flow, as well as the interaction of the liquid jet with the walls of the nozzle channel and the surrounding gaseous environment. The complex action of these factors leads to the destruction of the flow into individual droplets. At the same time, the contact surface of the fuel with the heated air charge increases significantly, which accelerates the processes of heating, evaporation, mixture formation and combustion.

For liquid injectors, atomization quality depends on the design of the nozzle, the properties of the fuel and the injection pressure. In addition, in diesel engines, fuel is injected into a combustion chamber of limited volume, in which the air charge has a fairly high temperature and is under a pressure of 4...16 MPa. These are additional factors influencing the quality of liquid fuel atomization.

The atomization process occurs in several stages and is characterized by a number of parameters, which can be divided into macroscopic and microscopic. Macroscopic parameters include parameters that characterize the geometric dimensions and shape of aerosol formations formed during the process of fuel atomization and are called *atomization plume.* Microscopic parameters include parameters characterizing the size of aerosol droplets, their homogeneity, and the distribution of aerosol droplets in the plume by size and speed.

5.9.2 Stages and macroscopic parameters of the injection process, their influence on mixture formation in diesel engines

In Fig. 5.175 shows eight frames of high-speed filming, which show the development of a atomization plume of light diesel fuel, filmed within one injection. The injection conditions are as close as possible to the conditions characteristic of a high-speed engine. In the case under consideration, fuel atomization was carried out in an experimental chamber filled with nitrogen through a fuel injector hole with a diameter 0.4 mm at a pressure in front of the nozzle of 70 MPa and a gaseous medium density of 25 kg/m^3. Nitrogen was used to fill the chamber to prevent self-ignition of the fuel, which distorts the pattern of atomization plume development. To analyze images, there are

the cone opening angle.

Numerous studies have shown a strong dependence of the macroparameters of the atomization process on the injection pressure and the amount of back pressure in the combustion chamber. In Fig. 5.179 presents experimental data demonstrating the nature of changes in the macroparameters of the atomization plume for various injection pressures and back pressures in the combustion chamber when using light and heavy fuel.

Analysis of the dependencies shown in Fig. 5.179 shows that with increasing injection pressure, the range of the atomization plume increases in all cases. This is explained by an increase in the speed of liquid flow from the fuel injector hole, which largely depends on the pressure in front of the nozzle.

It should be noted that, all other things being equal, the macroparameters of fuel-air mixing of heavy fuel are worse than for light fuel.

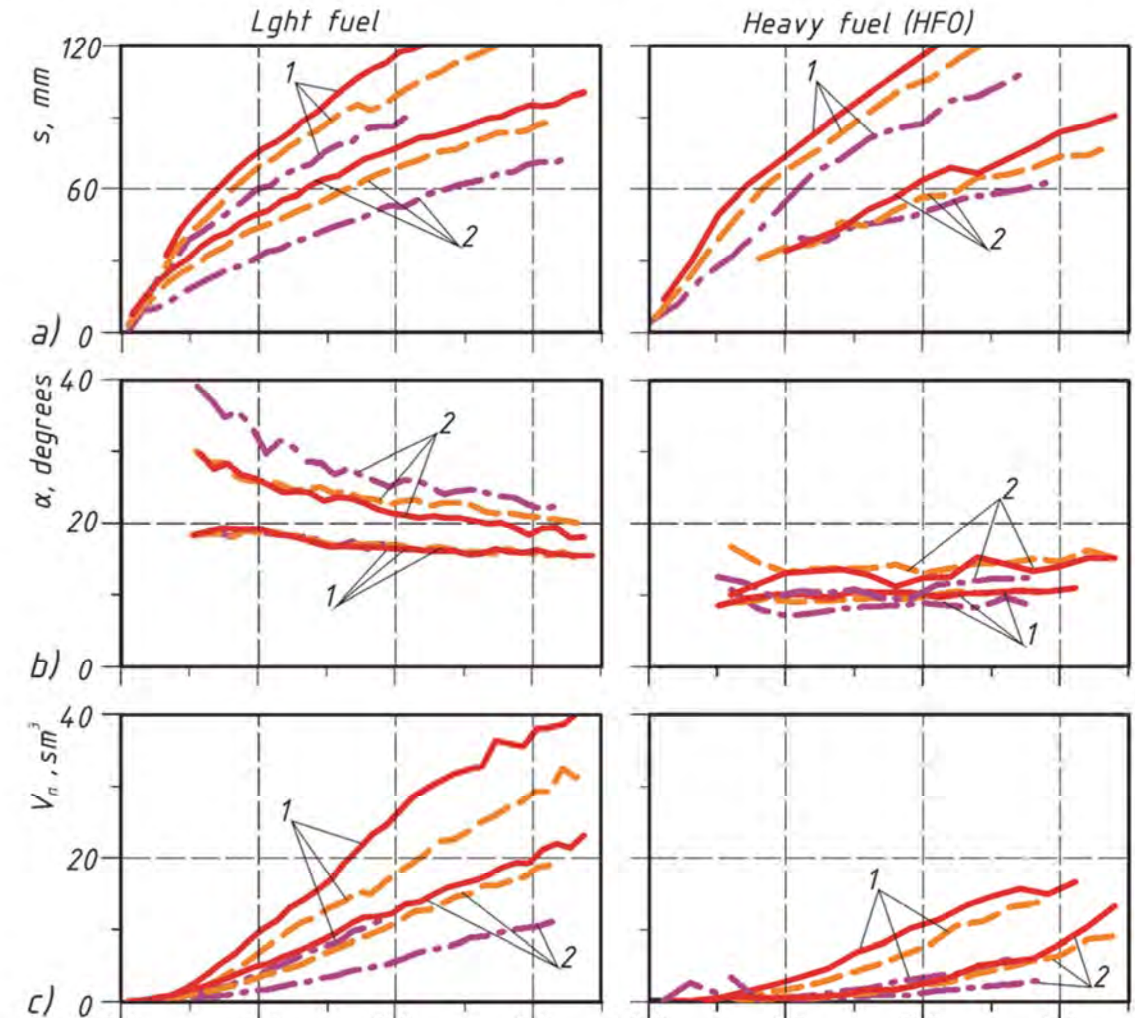
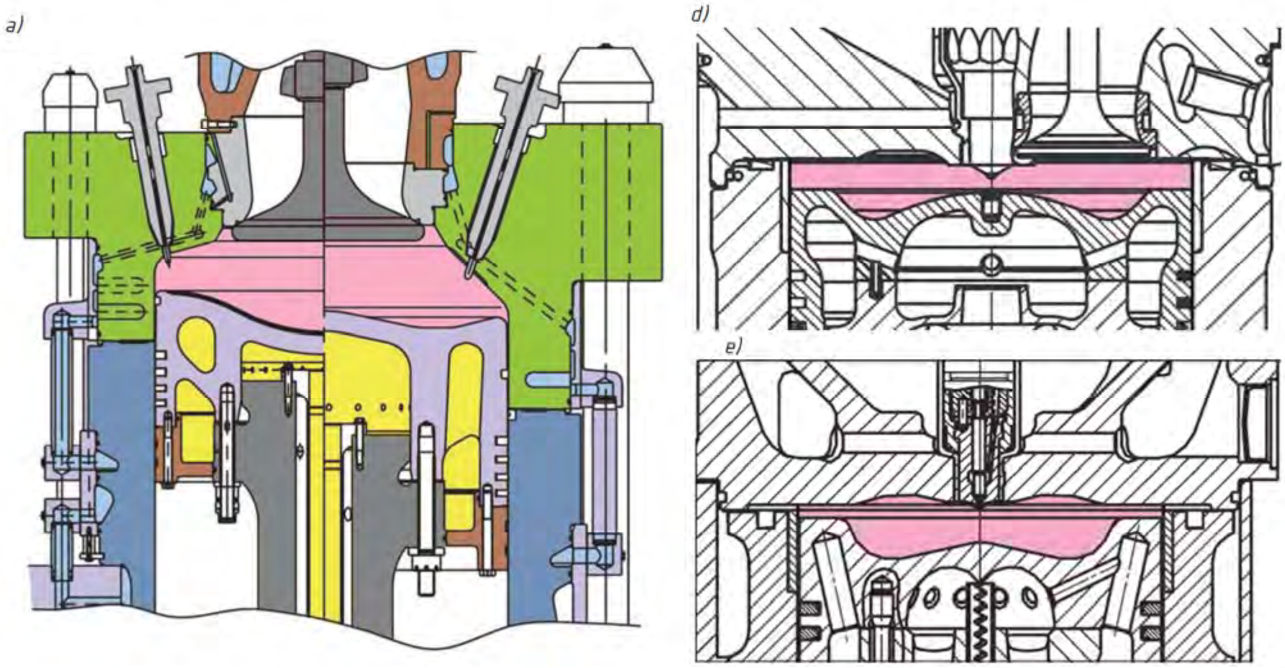


Figure 5.179 – Change in time of macroparameters of the atomization plume depending on the injection pressure for light and heavy fuel: 1 – back pressure 1.4 MPa; 2 – back pressure 5.0 MPa; *a* – depth of penetration of the atomization plume (*s*); *b* – taper of the atomization plume (α); *c* – volume of space coverage by the atomization plume (V_{cc}): ——— – injection pressure 140 MPa; – – – – injection pressure 100 MPa; – · – · – – injection pressure 60 MPa (adapted from [111])

It is clearly seen that the magnitude of the backpressure has a significant influence on the macroparameters of the injection process. If for light fuel at low back pressures the taper value is practically independent of the injection pressure, then with an increase in back pressure there is a decrease in the taper angle with increasing injection pressure. For heavy fuels, the taper angle is

formed in the central part of the combustion chamber, in which the air is not used efficiently. To reduce the volume of the uncovered zone, a displacer is placed in the central part (Fig. 5.183 *a*, right), which increases the concentration of air in the penetration zone of the atomization plumes and reduces direct contact of fuel with the piston. This design was called Oros, which translated from Greek means hill, small slide.

On medium-speed and some high-speed marine engines, of all the variety of forms, the most widely used is the «Hesselmann» type chamber, which maximally follows the contours of the atomization plume with a central location of the multi-hole fuel injector (Fig. 5.184 *b*). Such combustion chambers are located primarily in the piston, but a small part of the volume can be placed in the cylinder cover (Fig. 5.184 *c*). The central displacer on the bottom of the piston helps push air from areas not covered by plumes into the area of their formation and penetration, thereby improving the use of oxygen contained in the air.



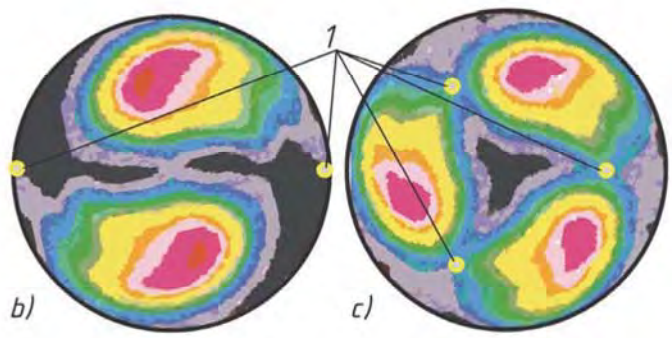


Figure 5.183 – Combustion chambers of marine diesel engines: *a* – versions of combustion chambers with a spherical recess in the piston and an Oros type piston bottom shape; 1 – fuel nozzle tips; *b*, *c* – distribution of fuel and local temperatures throughout the volume of the combustion chamber when using two and three peripheral injectors per cylinder; *d* – combustion chamber of the Hesselmann type of a four-stroke diesel engine made in the piston; *e* – hesselmann type combustion chamber of a four-stroke diesel engine, made in the piston and partially in the cylinder cover (adapted from [22, 112])

The quality of volumetric mixture formation significantly depends on the presence and degree of air charge turbulization. In two-stroke diesel engines, the vortex motion of the charge is additionally created at the purge stage due to the inclined arrangement of the purge ports in the cylinder liner. For each specific case, there are optimal speeds and directions of charge movement, when deviating from which aerosol droplets and fuel vapors from the volume of one atomization plume can be transferred by the movement of the charge into the volume of another atomization plume, leading to the formation of local zones with a low fuel-air ratio and, as a result, deterioration mixture formation.

rower the range of droplet diameters in which the characteristic is located, the finer and more uniform the atomization it displays.

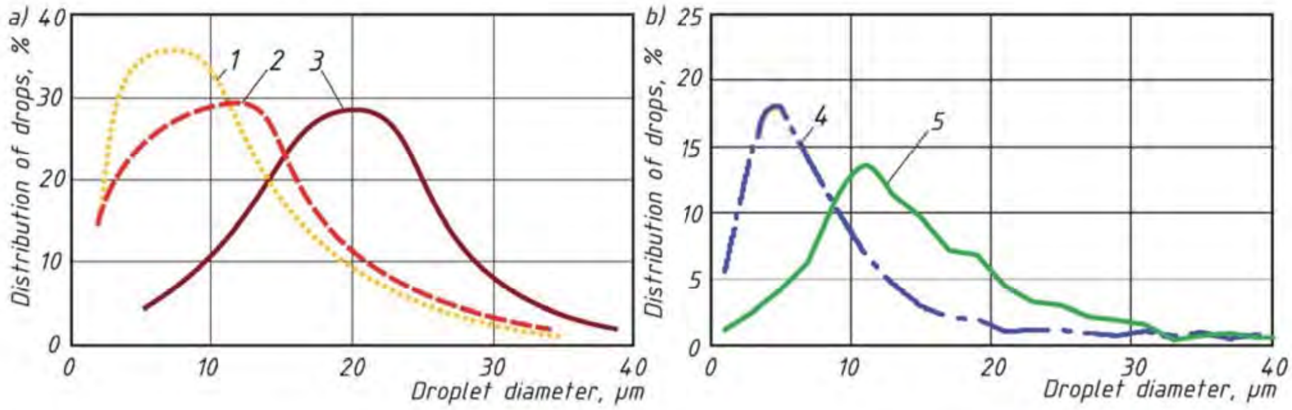


Figure 5.186 – Characteristics of fineness and uniformity of fuel atomization: *a* – for one total flow area with different diameters and numbers of nozzle holes; 1 – four nozzle holes with diameter 0.4 mm; 2 – two nozzle holes with diameter 0.57 mm; 3 – one nozzle hole with diameter 0.8 mm; *b* – under the same initial conditions for heavy and light motor fuel (nozzle hole 0.27 mm, pressure in front of the fuel injector 100 MPa, back pressure 1.4 MPa); 4 – heavy fuel (12 cSt); 5 – light fuel (3 cSt) (adapted from [9])

It should be understood that the average diameter of aerosol droplets is a very conditional parameter, since under the influence of a number of factors it can change both in time and depending on the location of its measurement along the length of the plume.

In table 5.9 presents the results of measurements d_{32} under various injection conditions, obtained for a accumulator-powered fuel supply system with a accumulator located in the injector housing.

The injection was controlled using an electronic system. The injection duration in all cases was 5 ms.

Table 5.9 – Results of measurements of the average droplet diameter (d_{32}) of light motor fuel under various injection conditions

Diameter holes, mm	Pressure in front of the nozzle, MPa	Density gas environment, kg/m^3	Localization measurements, mm	Average diameter according to Sauter, µm
0.4	83	25	120	25.40
0.2	83	25	120	23.28
0.4	28	25	120	39.19
0.4	55	25	120	27.10
0.2	83	1.2	120	12.67
0.2	83	25	60	12.39
0.2	83	25	80	16.33
0.2	83	25	100	20.96

The patterns of particle formation and distribution are the result of the combined action of several factors, which together determine the characteristics of an aerosol. The main factors include: injection pressure, distance from the tip of the nozzle to the measurement localization, density of the gas medium, diameter and length of the nozzle hole of the nozzle, needle lifting speed, etc.

Changes in the average size and velocities of aerosol droplets over time. In Fig. 5.187 shows the time dependence of the change in the average diameter and the velocity distribution of aerosol droplets in the centre of the atomization plume during the injection of light fuel by the accumulator fuel supply system of a high-speed diesel engine. The dependences were obtained for dislocation measurements 20 mm from the fuel injector hole by processing 20 thousand measurements obtained over 200 consecutive injections.

ters and number of nozzle hole in the fuel nozzle tips. As experimental data show, with a decrease in diameter and an increase in the number of hole, other things being equal, the quality of atomization improves.

From Figure 5.186 and it can be seen that as d_p increases, the spraying becomes coarser and less uniform. The taper of the atomization plume increases with increasing diameter of the nozzle holes.

The first circumstance is explained by the fact that, as the diameter of the nozzle holes increases, more time is required for the disintegration of the fuel jet, since the disintegration of the liquid core begins later in time and at a greater distance from the tip of the nozzle. In addition, as d p increases, the role of initial disturbances decreases.

The second circumstance is explained by the fact that as the diameter of the nozzle hole opening increases, the cone angle of the atomization plume decreases, since the length of the fuel atomization plume increases approximately in proportion to the square root of the nozzle hole diameter (5.1, 5.2), and the fuel supply is proportional to the square of the diameter. As a result, the diameter of the atomization plume increases more slowly than its length (Fig. 5.192), and the length of the atomization plume increases.

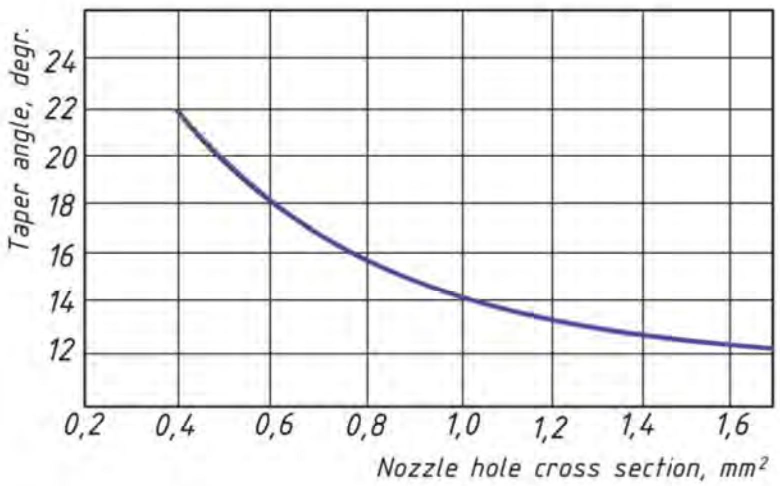


Figure 5.192 – Change in the cone angle of the atomization plume as a function of the diameter of the fuel injector opening of the nozzle (adapted from [9])

In practice, based on the above circumstances, the selection of hole diameters and their number is one of the most significant factors in optimizing mixture formation in diesel engines, since the task of mixture formation is not only to obtain a fine aerosol, but also its uniform distribution throughout the volume of the combustion chamber.

If the diameter of the nozzle holes is too small, the range of the atomization plume decreases, and small drops, having a low impulse, quickly lose their speed and concentrate around the nozzle, where there is not enough air for complete combustion of the fuel. At the same time, air that is not involved in the combustion process remains at the periphery of the chamber. The situation is further aggravated by the fact that an increase in the conicity of the atomization plumes from several peripheral holes leads to the interpenetration of the aerosol of one jet into another (Fig. 5.193 *a*). As a result, zones are formed that are extremely over-enriched in fuel, in which efficient combustion is simply impossible. The appearance of overlapping zones of plumes can also be caused by the mobility of the charge in the combustion chamber, especially when the injectors are off-center. In this regard, when optimizing mixture formation processes, it is necessary to take into account the influence on this process of the degree and nature of turbulization of the air charge in the combustion chamber.

As the nozzle holes diameter increases, the atomization quality deteriorates. But larger drops, having greater momentum, reach the periphery of the combustion chamber. This ensures a more uniform distribution of fuel throughout its entire volume, which leads to more efficient combustion (Fig. 5.193 *b*).

If the diameter of the nozzle holes is too large, an increase in the atomization plume range can lead to some of the droplets falling onto the relatively cold walls of the cylinder liner, as a result of which the combustion quality will significantly deteriorate (Fig. 5.193 *c*).

where $dh_p/d\varphi$ – speed of lifting the plunger along the angle of rotation of the crankshaft (m/s); $d\varphi/d\tau$ – angular velocity (rad/s).

At constant speed n angular velocity (rad/s)

$$\omega = \frac{d\varphi}{d\tau} = \frac{\pi n}{30}. \tag{5.10}$$

Taking into account expressions (5.9) and (5.10), equation (5.8) of the volumetric flow of the fuel pump (m^3/s) takes the form:

$$G_p = f_p \frac{dh_p}{d\varphi} \frac{\pi n}{30}. \tag{5.11}$$

Equating the right-hand sides of equations (5.5) and (5.11) based on the initial position that the volumetric fuel flow rate of the fuel injector is equal to the volumetric flow of the pump ($G_i = G_p$) and, solving the equality with respect to p_{inj} (Pa), we obtain:

$$p_{inj} = p + \frac{\rho_f \pi^2}{1800} \left(\frac{f_p}{\mu_n f_n} \right)^2 \left(\frac{dh_p}{d\varphi} \right)^2 n^2. \tag{5.12}$$

Analysis of equation (5.12) shows that the injection pressure depends on the density of the fuel, the ratio of the area of the fuel pump plunger to the effective flow area of the fuel injectors $f_p/(\mu_c f_c)$ the plunger lifting speed at the considered moment of injection $dh_p/d\varphi$ and rotation speed p diesel.

Due to the assumptions made, equation (5.12) allows us to estimate the nature of the influence of the main factors on the injection pressure with a certain degree of accuracy. In real conditions, at pressures typical of marine diesel fuel systems, liquid fuel behaves like an elastic medium that contracts under the influence of force and expands when this force is removed.

If a disturbance is created in the volume of liquid, it will propagate in the form of a pressure wave at the speed of sound (for fuel on average 1450 m/s). When the wave encounters an obstacle, it is partially or completely reflected from it, causing a reverse pressure wave to appear. As a result of the disturbances created by the plunger, direct pressure waves occur in the pipeline. Reflecting from obstacles, which can be shut-off valves, injector needle valves, fuel injector openings, etc., reverse pressure waves arise in the high-pressure line. Overlapping each other, pressure waves create a complex picture of processes, the nature of which largely determines the efficiency of fuel atomization. In particular, under the influence of pressure waves arising from the impact of the mass of fuel on the closed needle valve of the injector or the reverse wave from the closed injection valve of the injection pump, so-called after-injections can be observed – additional lifts of the needle after the end of the main injection. During the injection period, the fuel pressure slightly exceeds the pressure at which the injector starts feeding, so the atomization quality is very low. The fuel, flowing out of the nozzle holes, partially settles on the tip of the nozzle, causing severe carbon deposits. In addition, fuel injection occurs, as a rule, on the expansion line, which leads to an increase in the afterburning period and a decrease in diesel efficiency.

The occurrence of after-injections can be either a consequence of incomplete refinement of fuel equipment at the factory, or the result of changes in technical condition during operation. The occurrence of after-injections is most likely in a fuel supply system with a long discharge pipeline; it is for this reason that they try to place fuel pumps as close to the injectors as possible

When designing fuel equipment, complex hydrodynamic calculation methods are used that take into account the viscosity of the fuel and its compressibility, the volume and length of the injection pipelines, the presence of fuel leaks in the fuel pump and injector, the influence of direct and reflected waves on the nature of pressure changes in various sections. Three-dimensional computer modelling methods are being widely introduced into design practice, allowing not only to obtain numerical solutions, but also to visualize the processes occurring in fuel systems.

draulic cylinder, and for systems with a mechanical drive – on the profile of the cam washer of the fuel pump.

With a mechanical drive of the injection pump, the choice of the fuel cam profile is carried out on the basis of calculations of the main geometric dimensions of the fuel equipment and the kinematic characteristics of the fuel pump plunger (the average speed of the plunger c_m on the interval of the geometric useful stroke of the plunger).

The main parameters include those that provide the specified characteristics of fuel injection over the duration of supply: the geometric dimensions of the working profile of the fuel cam, the lift angle and the amount of full rise of the profile.

In marine diesel engines, fuel cam profiles are most often used, defining trapezoidal and triangular or similar laws for changing the speed of the plunger depending on the angle of rotation of the cam shaft (Fig. 5.197).

The first of these profiles (Fig. 5.197 *a*) is characterized by a constant plunger speed during fuel injection, which creates certain convenience when regulating the fuel equipment on the engine according to injection advance. The second (Fig. 5.197 *b*) allows you to obtain the highest average speed of the plunger in the section of its active stroke and make the most of the given full rise of the fuel cam profile.

Quite often there are cases when only a section of the ascending velocity branch is used as a working one (Fig. 5.197, *c*).

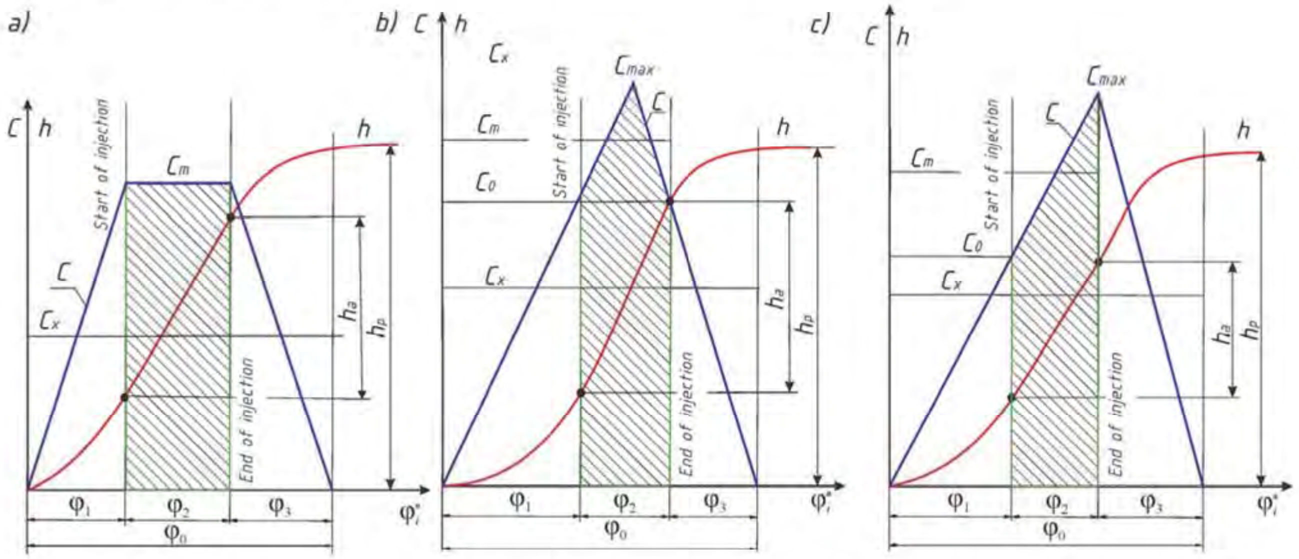


Figure 5.197 – Typical characteristics of fuel cam profiles: *a* – trapezoidal law of change in plunger speed; *b* – triangular law with equal plunger speeds at the beginning and end of injection; *c* – triangular law using the section of the ascending velocity branch; *c* – plunger speed; *h* – plunger stroke; φ – angle of rotation of the cam shaft; $c_{\max}$ – maximum plunger speed; c_m – average speed in the section of the active stroke of the plunger h_a; c_0 – speed corresponding to the beginning of the geometric active stroke of the plunger (start of supply); c_x – average speed over the period of full stroke of the plunger h_p; h_p – full stroke of the plunger (raising the profile of the fuel cam); h_a – active stroke of the plunger; φ_0 – fuel cam profile elevation angle; φ_1 – duration of the plunger lifting phase until the geometric start of fuel supply; φ_2 – geometric injection duration; φ_3 – duration of the fuel cut-off phase (adapted from [9])

The average plunger speeds for fuel cam profiles with a triangular law of speed change, all other things being equal, are 6...12% higher.

The permissible acceleration of the plunger usually lies in the range of 200...400 m/s^2, and in some cases it can reach 500 m/s^2 or more.

The acceleration value is the starting point for choosing a plunger spring, which must ensure constant contact of the pusher roller with the fuel cam profile.

In practice, quite often, to ensure the specified injection parameters, asymmetric laws for changing the speed of the plunger are used, in which the highest speed is achieved in the section when the

the rate of heat release $dQ/d\varphi = 0$. This period is characterized by the absence of fuel supply, a decrease in pressure in the cylinder, a further decrease in the oxygen concentration and an increase in the concentration of the final combustion products. For some high-speed diesel engines, this period may last until the exhaust valve opens, which is an extremely undesirable phenomenon. It leads to an increase in heat losses and an increase in the thermal stress of the parts of the cylinder-piston group, and, consequently, to a deterioration in the efficiency of the diesel engine.

In new Miller cycle engines, in order to limit the maximum cycle pressure p_z, most of the combustion process is transferred to the expansion line. As a result, the pressure and temperature of the gases at the end of expansion are higher in these engines. Losses associated with higher gas parameters at the end of expansion in the piston part are compensated by their subsequent expansion in a highly efficient turbocharger unit. This improves the performance of the charging system, which is a key requirement for an efficient Miller cycle.

To analyze the nature of the combustion process, a number of indicators and characteristics are used, which will be discussed below.

Fuel combustion is the main process of the operating cycle, during which heat is released, which is converted in the engine into useful mechanical work. When assessing combustion processes, intermediate physicochemical changes in the working fluid are not considered, but only the final results of chemical reactions of combustible fuel components with atmospheric oxygen are taken into account.

When assessing the parameters of the working fluid during the combustion process, the actual nature of the change in pressure in the cylinder is conditionally replaced by isochoric *cz* and isobaric *zz'* by heat supply processes (Fig. 5.201). This cycle is called a mixed heat input cycle or mixed cycle.

From the moment fuel begins to be supplied to the cylinder (point *h*, Fig. 5.201) until the end of combustion, complex physical and chemical processes of evaporation, ignition and combustion of fuel, heat release, changes in pressure and temperature occur. All these processes occur at finite speeds, which determine the duration of the process as a whole. For most engines, the combustion process occupies a cycle section of 20...35°.

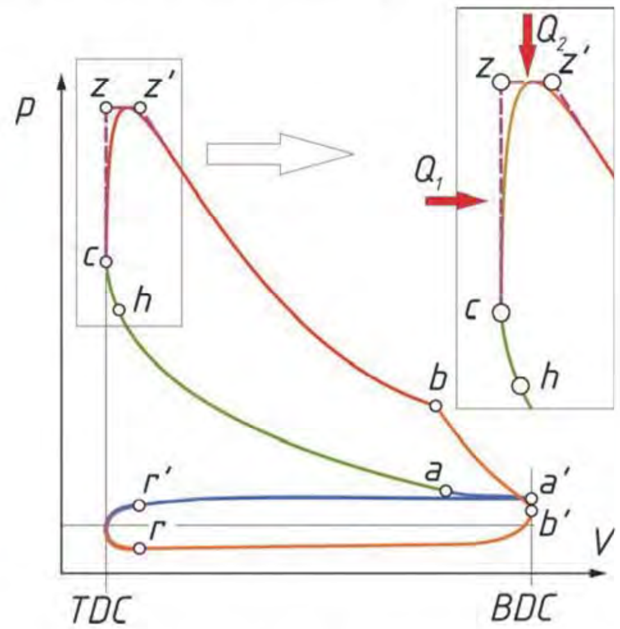


Figure 5.201 – Scheme of the combustion process in a cycle with mixed heat supply

Fuel injection begins a few degrees before TDC (point *h*, in Fig. 5.201). This makes it possible to distribute the fuel aerosol throughout the combustion chamber and ensure the occurrence of physico-chemical processes preceding combustion (pre-flame processes) before the piston reaches TDC.

The distribution of fuel droplets in the air charge environment ensures their rapid heating and evaporation, as well as direct contact of fuel vapor with air oxygen, which acts as an oxidizer.

$$\overline{c}''_{v_{z'}} = \frac{(mx_{z'} + \gamma_r)\ \overline{c}''_v + [\alpha(1+\gamma_r) - (x_{z'} + \gamma_r)]\ \overline{c}'_{v_c}}{\alpha\ (1+\gamma_r) + (m-1)\ x_{z'}}.$$

Pressure and temperatures p_c, $p_{z'}$, T_c and $T_{z'}$ are determined from the characteristic equations of state:

$$p_c V_c = 8{,}314\, M_c T_c,$$
$$p_{z'} V_{z'} = 8{,}314\, M_{z'} T_{z'},$$

where M_c is the number of moles of charge at the end of compression; $M_{z'}$ – number of moles of charge at the end of combustion.

Heat equivalent to the work of the isobaric process zz':

$$\begin{aligned}\Delta L_{zz'} &= p_{z'} V_{z'} - p_z V_z = p_{z'} V_{z'} - \lambda p_c V_c = 8{,}314 M_{z'} T_{z'} - 8{,}314 \lambda M_c T_c = \\ &= 8{,}314 \beta_{z'} L (1+\gamma_r)\ T_{z'} - 8{,}314 \lambda L (1+\gamma_r)\ T_c.\end{aligned} \quad (5.18)$$

In these equations, the quantities $V_{z'}$ and $T_{z'}$ remain unknown, which are determined from the combustion equation.

Substituting expressions (5.14), (5.17) and (5.18) into equation (5.13) and performing transformations, taking into account that the relationship between isobaric and isochoric heat capacities $\overline{c}''_{p_{z'}} = \overline{c}''_{v_{z'}} + 8{,}314$, we obtain the combustion equation as:

$$\frac{\xi_{z'} Q_l}{\alpha L_0} + [\overline{c}'_v + 8{,}314\lambda + \gamma_r (\overline{c}''_v + 8{,}314\lambda)]\ T_c = \beta_{z'} (1+\gamma_r)\ \overline{c}''_{p_{z'}} T_{z'}. \quad (5.19)$$

where L_0 is the theoretically required amount of dry atmospheric air to burn one kilogram of fuel, kmol/kg.

Equation (5.19) is usually solved by the method of successive approximations. As a first approximation, they are specified by the temperature value within the range $T_{z'}$ = 1700…2000 K.

As a result of substituting into equation (5.19) all the numerical values of the known quantities $\xi_{z'}$, Q_l, γ, L_0, $\overline{c}'_v$, $\overline{c}''_v$ and $\lambda = p_z / p_c$, as well as the dependence $\overline{c}''_{p_{z'}}$ on the desired temperature $T_{z'}$, the combustion equation is reduced to a quadratic equation type:

$$A T_{z'}^2 + B T_{z'} - C = 0;$$

$$T_{z'} = \frac{-B + \sqrt{B^2 + 4AC}}{2A}, \text{ K}.$$

The temperature $T_{z'}$ at which the parts of the equation converge is the desired one.

For marine diesel engines, the temperature values at the end of visible combustion $T_{z'}$ lie within the following limits: low-speed 1600…1800 K; medium speed 1700…1900 K; high-speed 1800… 2000 K.

The volume of the cylinder at the end of visible combustion is determined depending on the degree of pre-expansion: $V_{z'} = \rho V_s$.

The degree of preliminary expansion can be determined as a result of the joint solution of the equations of state of the gas at points z' and c. Dividing the characteristic equations of state term by term and transforming the resulting equality taking into account the relations $\lambda = p_{z'} / p_c$; $\rho = V_{z'} / V_c$; $\beta_z = M_{z'} / M_c$, we obtain a formula for calculating the degree of preliminary expansion: $\rho = \beta_z T_{z'} / (\lambda T_c)$. At nominal mode for most diesel engines ρ = 1.2…1.6. The degree of preliminary expansion depends on the method and quality of the mixture formation processes, fuel combustion, fuel injection advance angle, load and diesel engine speed.

Indicators of the dynamism of the operating process. To assess the nature of changes in loads on engine structural elements during fuel combustion, indicators of the dynamism (rigidity) of the operating process are usually used: the average and maximum rate of pressure increase.

The duration of the self-ignition delay period is mainly determined by the temperature of the charge at the time of fuel injection, the properties of the fuel itself, and the quality of its atomization. The latter largely depends on the performance of the fuel equipment.

To obtain a given pattern of pressure changes in the working cylinder, it is necessary to take into account the time required for pre-flame processes. To do this, the moment of the start of fuel supply is set earlier than the theoretically determined moment of the start of heat release by the auto-ignition delay angle (φ_{fa}). In practice, the influence of the self-ignition delay period on the operating process is taken into account by setting the static advance angle of the fuel pump supply φ_{pdt} (Fig. 5.196).

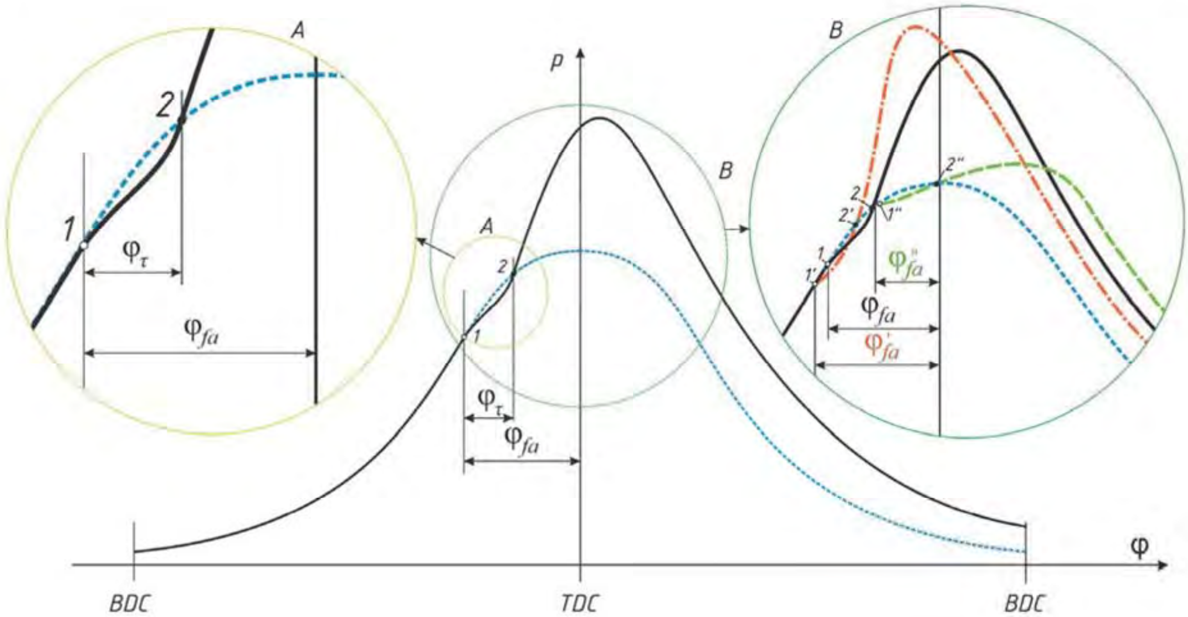


Figure 5.206 – Self-ignition delay period and its effect on the engine's operating process: 1 – the moment the supply starts; 2 – moment of separation of the combustion line from the compression line. – – – – – compression line; —— – combustion line at the optimal supply advance angle; – · – · – combustion line during early fuel injection; – – – – combustion line with late fuel injection (adapted from [9])

With an increase in φ_{pdt}, fuel is injected into the cylinder earlier (point 1′ in Fig. 5.206), which leads to its earlier ignition. As a result, a greater amount of heat is released even before the piston reaches TDC, which leads to a sharper increase in pressure and an increase in its maximum value. The operating process becomes more dynamic and more rigid. With a further increase in the advance angle, this tendency will weaken, since the fuel will be injected into an environment with a lower temperature and pressure, and this will lead to an increase in the auto-ignition delay period.

With an increase in φ_{pdt}, the efficiency of a diesel engine initially increases, since a certain increase in the compression work up to TDC is more than compensated by an increase in the thermal efficiency of the cycle due to the supply of heat to the working fluid at a higher temperature. At large values of the angle φ_{pdt}, the compression work increases significantly and becomes greater than the gain in thermal efficiency, so the diesel efficiency decreases.

With a decrease in the angle φ_{pdt}, especially to the values corresponding to the beginning of fuel combustion after TDC (point 1″ in Fig. 5.206), the mechanical stress of the engine decreases, but at the same time its efficiency decreases. The combustion of the main portion of the fuel shifts to the expansion line, which increases the temperature of the exhaust gases and the thermal stress of the parts of the cylinder-piston group.

Obviously, the injection advance angle must increase with increasing engine speed in order to provide the necessary time period for pre-flame processes to occur. In addition, changes in engine load boost pressure, external conditions, and fuel grade may require adjustments to the supply advance angle.

The ignition and combustion of the first portions leads to the fact that subsequent fuel, despite the poorer atomization quality, enters an environment with elevated temperature and pressure, in which there are already pockets of open flame. This promotes rapid evaporation of subsequent portions of fuel and intense combustion. This organization of the operating process led to an increase in engine efficiency. However, the concentration of the entire combustion process in a narrow section of the cycle has led to an increase in combustion severity, an increase in the noise level generated by the engine, and an increase in loads on the parts of the cylinder-piston group and the crankshaft.

The second version of the supply law with constant injection pressure is inherent in accumulator systems with a large accumulator volume, when the loss of a portion of fuel equal to the cyclic supply does not lead to a significant drop in pressure in the system (Fig. 5.210 *b*). This method is characterized by the fact that throughout the entire injection the quality of atomization remains consistently high, as a result of which pre-flame processes and subsequent combustion proceed quite quickly. As in the first case, with such an organization of the injection process, high engine efficiency is achieved, but large thermal and mechanical loads arise on the CPG and crankshaft.

To avoid the noted phenomena in a number of modes, when the time factor allotted for fuel combustion is not so critical (for example, in LSE), a gentle law of increase in injection pressure is used in the initial stage (Fig. 5.210 *c*). This supply law is implemented by transferring the active stroke of the plunger to a section with lower acceleration, for example, by changing the valve opening angles of valve-type pumps or by changing the position of the pusher roller relative to the fuel injection pump drive cam. This law is most simply implemented in engines with hydraulically driven fuel pumps or in accumulator injection systems with electronic fuel control. For example, in Mitsubishi UEC-Eco series engines, the use of such a law is considered as a compromise solution between the efficient and environmental performance of the engine when operating at low speeds, but with a high load. The influence of the supply law with a gentle increase in pressure on the operating process of a low-speed engine of the ME series from MAN is shown in Fig. 5.210 *a*.

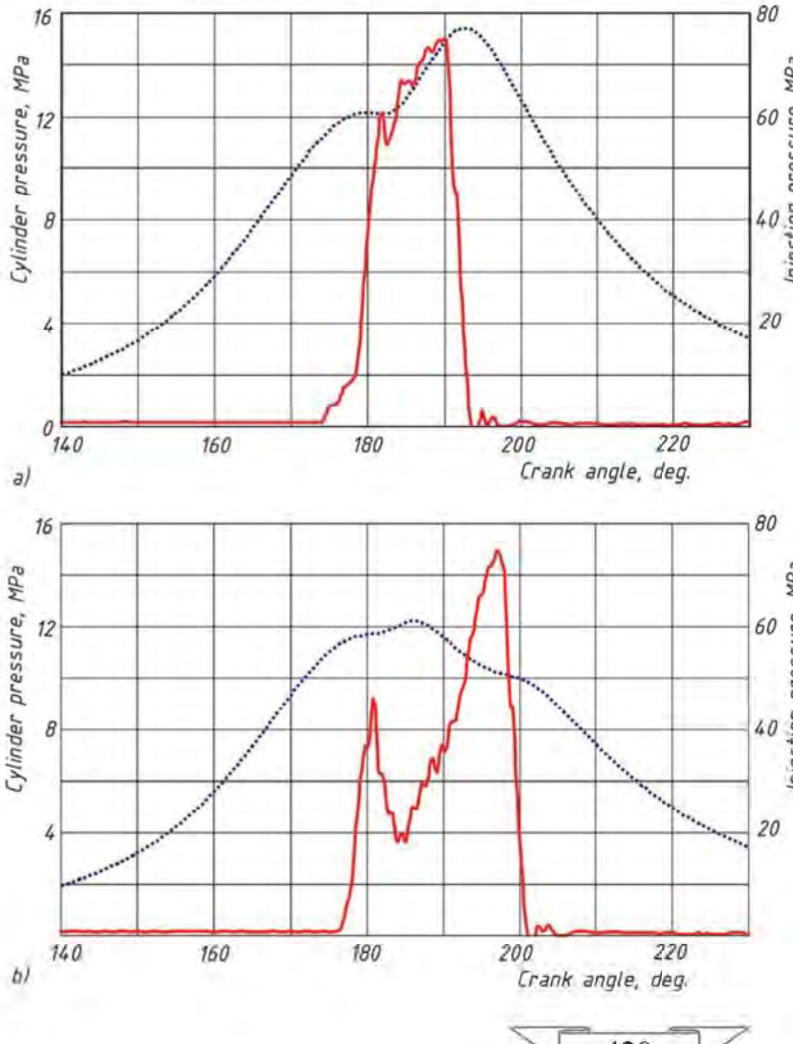


Figure 5.211 – Operating process of a low-speed engine with a gentle increase in injection pressure in the initial stage (*a*) and with two-phase fuel injection (*b*): —— fuel pressure in front of the fuel injector; - - - – cylinder pressure (adapted from [115])

To reduce exhaust smoke, some manufacturers resort to supplying a small additional portion of fuel at the end of combustion, when the main supply has already completed (Fig. 5.210 *e*). The combustion of a fresh portion of fuel intensifies the afterburning processes of the remaining free carbon particles and thereby reduces their content in the exhaust gases. A three-phase fuel supply process with pre-injection and post-injection is also considered promising (Fig. 5.210 *f*). Some engines use not one, but several pre-injections, which makes it possible to find a reasonable compromise between the efficient and environmental performance of a diesel engine (Fig. 5.210 *g*).

Multiphase injection is most simply implemented in accumulator injection systemsor in systemswith hydraulically driven plungers. Most often, the pilot portion is formed by submitting a short control signal to an electrically controlled fuel injector, however, cases of spool control of two-phase supply are also known. Thus, in its high-speed diesel engines of the 3126 series, Caterpillar used a plunger design with a spool edge to interrupt the flow (Fig. 5.216).

When the plunger moves downwards under the action of the hydraulic piston, fuel is pumped through the fuel injector (Fig. 5.216 *a*). This continues until the spool groove on the plunger, connected to the discharge cavity by three vertical channels (Fig. 5.216 *d*), coincides with the discharge hole (Fig. 5.216 *b*). As a result, fuel injection stops until the upper edge of the spool bore closes the unloading hole (Fig. 5.216 *c*), after which injection resumes.

Thus, when the plunger moves downwards, two successive injections occur, the first is short, the value of which is determined by the position of the spool groove on the plunger, and the second is the main one, the value of which is determined by the value of the active stroke of the plunger. The interval between injections depends on the width of the spool bore. The size of the pilot portion in this pump-injector design remains constant in all modes, and the engine is controlled by changing the value of the main supply by the electronic engine control system.

In addition to dividing the cyclic portion into several sequential injections, parallel supply of pre- and main portions of fuel is used in marine diesel engines (Fig. 5.210 *h*).

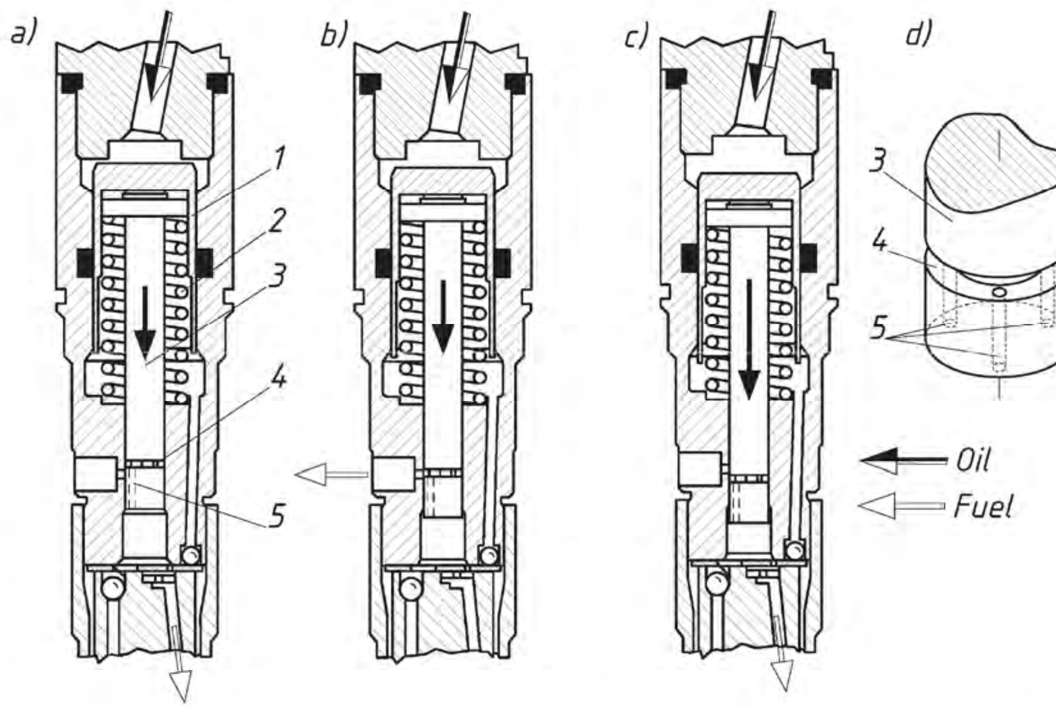


Figure 5.216 – Pump-injector with a hydraulic drive of a Caterpillar 3126 series engine with spool control of two-phase injection: *a* – the beginning of the injection of the pilot portion; *b* – pressure release; *c* – injection of the main portion; *d* – lower part of the plunger with a spool edge; 1 – hydraulic piston; 2 – return spring; 3 – plunger; 4 – spool groove; 5 – fuel bypass connecting channels (adapted from [44, 45])

This supply split is used in Wärtsilä 46C series engines. To implement this method, two injectors are installed on each engine cylinder, a peripheral one for injection of the pilot portion of fuel and a

Air compressed and heated to 120...130°C from the turbocharger passes through a humidifying column, in which it is cooled and humidified. In this case, the relative humidity of the air entering the engine can be maintained at a constant level of 99%.

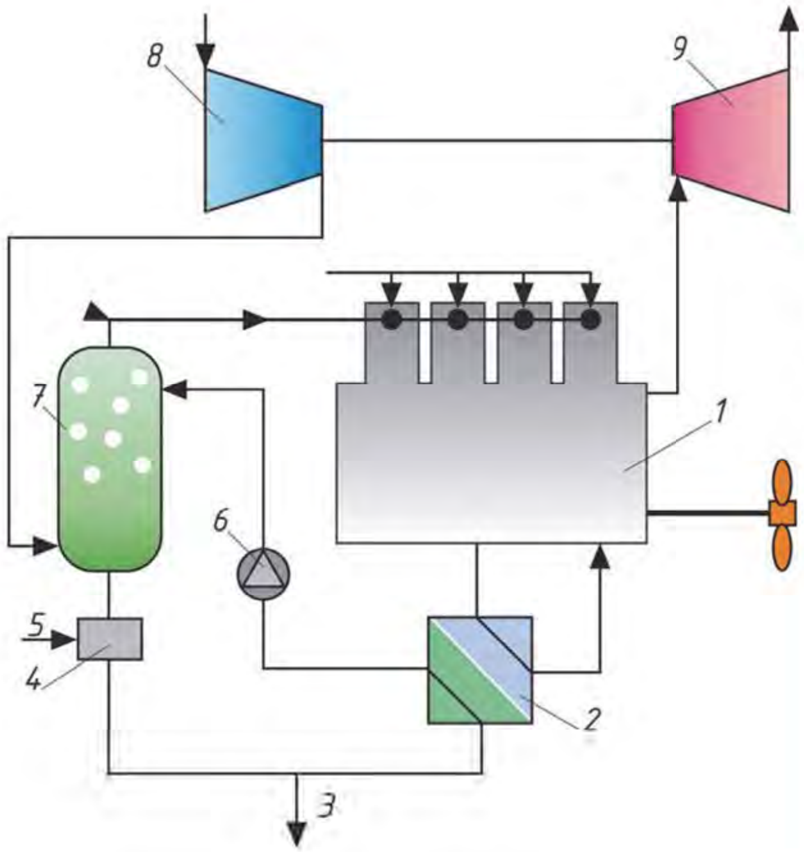


Figure 5.220 – Diagram of the installation for air humidification at the inlet to the engine from Munters Euroform: 1 – engine; 2 – heat exchanger; 3 – water drain; 4 – water tank; 5 – water supply; 6 – circulation pump; 7 – humidifying column; 8 – supercharging compressor; 9 – turbocharger turbine (adapted from [121])

Testing of the system showed that in operating mode the NO_x content decreased by 70...80%. Researchers explain this by the fact that the increased vapor content in charge air reduces temperature peaks in the combustion chamber.

Wärtsilä company in its early developments used a device for saturating the air entering the combustion zone with water vapor (Combustion Air Saturation System (CASS)) from Marioff Oy company. A water aerosol was introduced through nozzles directly into the charge air flow immediately after the turbocharger (Fig. 5.221).

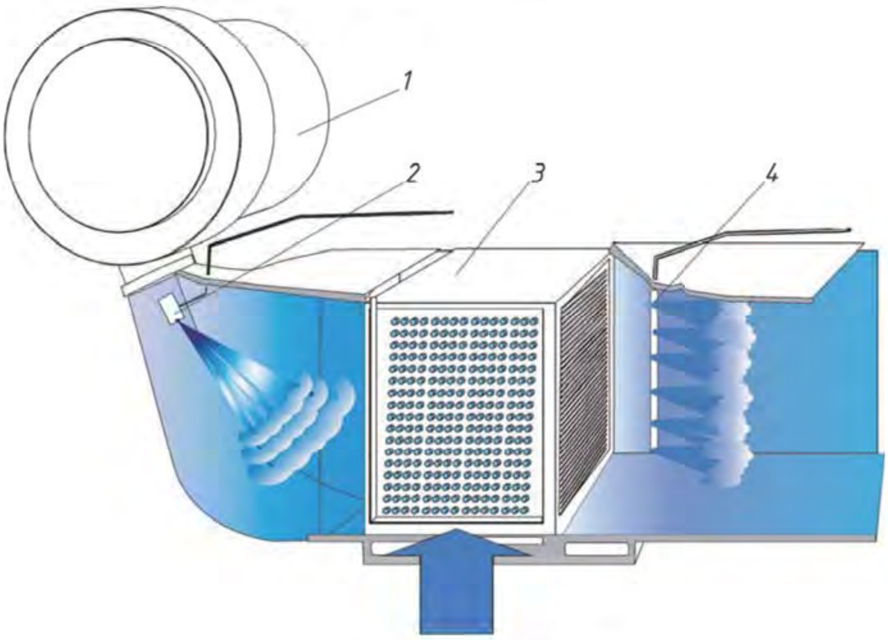


Figure 5.221 – Wärtsilä air humidification installation: 1 – turbocharger; 2 – injector for water injection; 3 – air cooler; 4 – additional water injection (adapted from [122])

For high-speed engines, stratified systemsare considered as promising injection fuel and water through one spray, equipped special spool valve device, allowing during periods between injections fill water part internal cavities, adjacent to fuel injector needle. The diagram of such a nozzle is shown in Fig. 5.227.

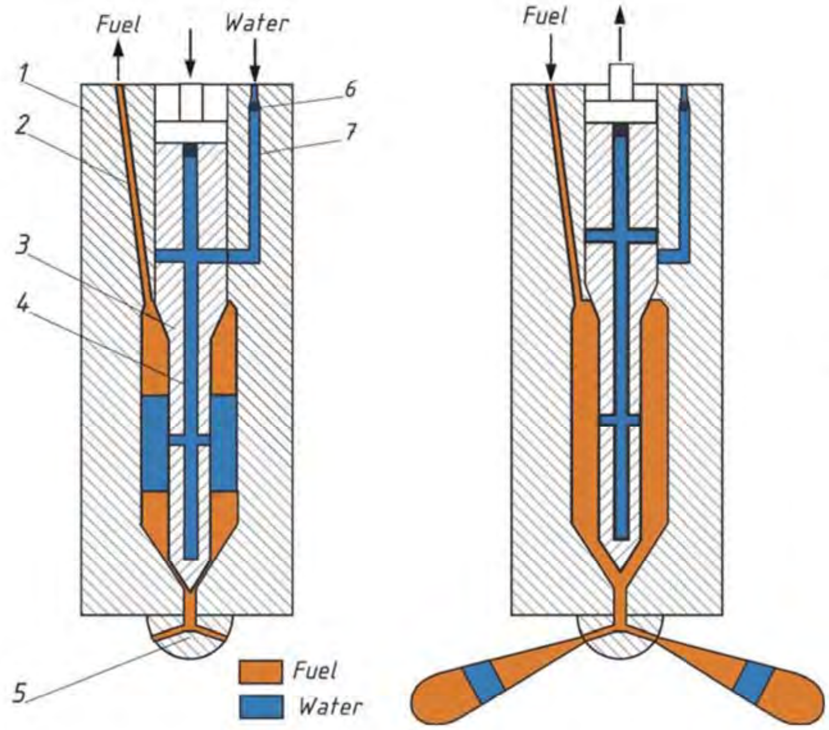


Figure 5.227 – Nozzle for stratified injection of water and fuel in a high-speed diesel engine: 1 – nozzle housing; 2 – fuel supply channel; 3 – needle valve; 4 – channel for supplying water to the cavity of the sprayer; 5 – fuel nozzle tip; 6 – check valve of the water line; 7 – water supply channel (adapted from [127])

When the needle valve is closed, the water supply channel in the nozzle housing coincides with the channel in the housing of the needle valve. The pressure of the supplied water is slightly higher than the residual pressure in the fuel line, so part of the water enters the the cavity of the sprayer, displacing the fuel and forming a water layer in the the cavity of the sprayer. When the fuel injection pump is pumping fuel, the water supply channel is closed by a check valve, and when the needle valve is raised, the water and fuel cavities are additionally disconnected due to the water supply channel being blocked in the nozzle housing of the needle valve generatrix.

As a result, clearly defined zones of fuel and water are formed in the atomization plume formed by the fuel injector.

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SECTION 6

Marine engine oil systems

6.1 Purpose and composition of the system

The engine oil system serves for:
- supplying oil to rubbing surfaces to reduce friction between them;
- removal of heat released as a result of friction;
- removal of wear products, soot and other foreign particles from the friction zone.

In addition, the oil circulating in the system is used as a coolant to remove heat from heated parts, for example from piston heads, and also as a working fluid in hydraulic drive systems, for example gas distribution valves, hydraulic thermal gap compensators, hydraulic drive of high-pressure fuel pumps, pump-injectors, etc.

Lubricant is supplied to cylinder liners, bearings of the crankshaft and camshafts, turbochargers, pumps, drive and intermediate gears, valve guides, pushers of fuel pumps and timing mechanisms, valve rods and rocker arms. The general diagram of the oil system of a medium-speed four-stroke engine is shown in Fig. 6.1.

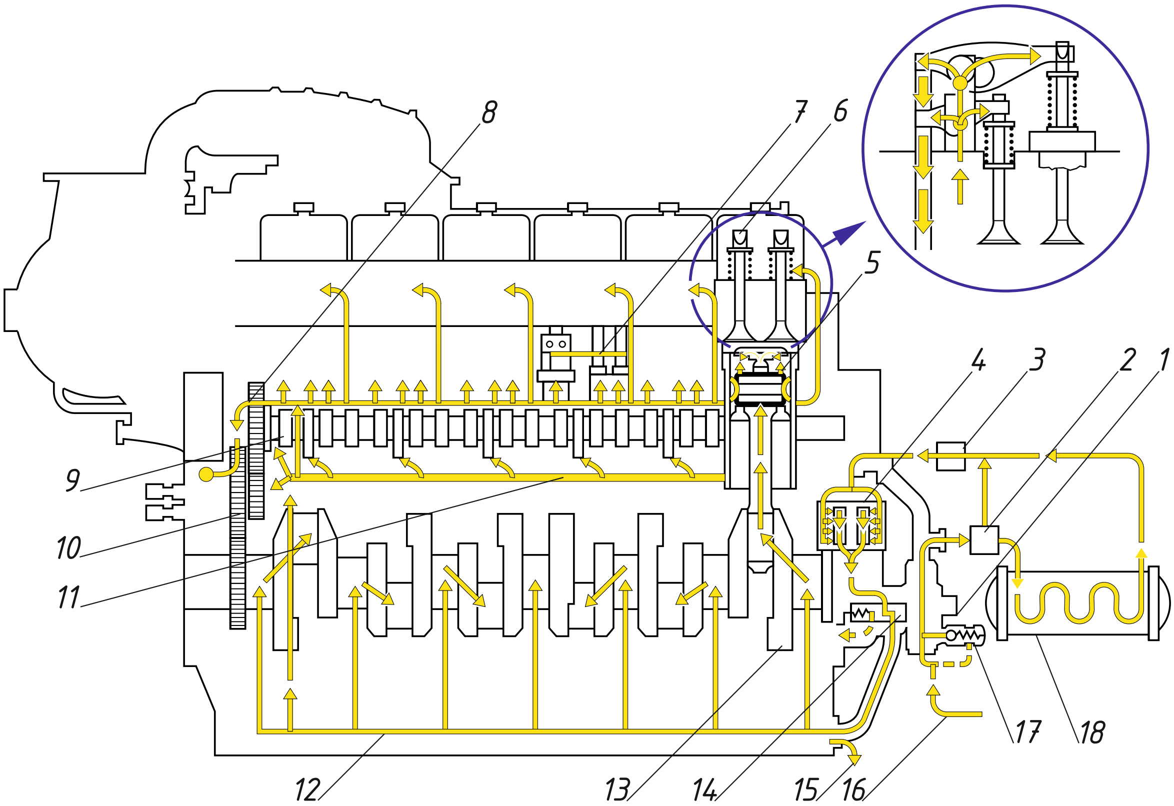


Figure 6.1 – Oil system of a four-stroke medium-speed engine TBD 645 from DEUTZ MWM: 1 – lubricating oil supply pump; 2 – lube oil thermostat; 3 – automatic filter (system side); 4 – backwash filter (engine side); 5 – pistons; 6 – valve stems; 7 – pushers valve and fuel pumps; 8 – line for supplying lubricant to valve pushers and fuel pumps; 9 – camshaft; 10 – intermediate gear; 11 – camshaft bearing lubrication line; 12 – distribution line in the oil pan; 13 – crankshaft; 14 – pressure control valve built into the backwash filter; 15 – draining lubricating oil into the tank; 16 – supply from the lubricating oil reservoir; 17 – safety valve; 18 – lubricating oil cooler (adapted from [1])

As follows from Fig. 6.1, the lubrication system includes oil tanks, oil lines, oil pumps, temperature and pressure regulators, oil filters, oil coolers (heaters).

Oil tanks are used for taking on board and storing oil reserves. Oil pumps are used to supply the required amount of oil to the discharge pipeline. Oil filters are used to clean oil from matter (wear

The possibility of exposing the oil receiver when rocking the engine and the high probability of oil foaming as a result of it getting on the moving parts of the crank mechanism are the main disadvantages of dry sump systems. In addition, such engines have a greater distance from the bottom point to the axis of the crankshaft, which in some cases complicates their arrangement with other elements of the vessel's propulsion complex. In a wet sump, the oil is constantly in contact with the products of incomplete combustion that enter the crankcase through leaks in the piston rings, which leads to more intensive aging of the oil (changes in its chemical composition and physical properties).

Systems with a «dry sump» do not have the disadvantages described above, in which the oil spent in friction pairs is pumped out of the sump by a special pump (or drained) into a separate tank, where its main supply is stored. It is precisely such systemsthat have become predominantly widespread on high-power marine engines.

An important indicator of circulating oil systemsis the circulation rate, that is, the ratio of the hourly supply of circulation pumps to the volume of oil in the system. This indicator significantly affects the service life of the oil and the frequency of its replacement. Increasing the circulation rate by half reduces the oil service life by 2...4 times. For high-speed engines with a dry sump this figure is 40...60, and for high-speed engines with a wet sump it is 50...100. In medium-speed high-power engines with a dry sump, the circulation ratio is 12...30. The use of large-volume circulating oil storage tanks in dry-sump circulation systemsallows low-speed engines to provide a circulation ratio of 4…15. For this reason, the service life of the oil in low-speed engines, with regular cleaning, can reach tens of thousands of hours, and is sometimes comparable to the service life of the engine itself.

From the reserve storage tank, the circulating oil is sucked in by an oil pump through a special oil intake pipe, equipped at the inlet with a coarse filter in the form of a large metal mesh. The purpose of the filter is to prevent large objects from entering the oil pump. In some cases, to perform a similar function, a separate coarse oil filter is installed in front of the pump inlet.

Screw pumps with three working propellers are used as an oil pump on large marine engines (Fig. 6.3 *a*). On smaller engines, gear-type pumps with two (Fig. 6.3 *b*) or three working gears are predominantly used.

The performance of the pumps is selected in such a way that the oil supply covers all needs, both for lubrication of friction pairs and for cooling heated parts (for example, cooling pistons in low-speed engines), as well as for hydraulic drive of mechanisms using circulating oil as a working fluid (for example, reverse mechanism for the pump block of the RTA series engines from Sulzer (Fig. 5.38)). In this case, the oil consumption for cooling and hydraulic drive can be several times higher than the consumption associated with lubrication of engine friction pairs. Considering that during operation, oil consumption may increase, pump performance is taken with a significant margin.

Pumps in large diesel engines are usually driven by an electric motor, which allows maintaining stable operating parameters of the system regardless of the engine operating mode. In addition, this allows you to pump oil through the system when the engine is not running, for example before starting it, or for some time after stopping it to stabilize the temperatures of the cooled parts. For small engines, the oil pump is usually driven directly from the engine crankshaft through a gear transmission (Fig. 6.3 *b*). In this case, an additional bleed pump is provided, which serves to supply oil to the system when the engine is not running.

To limit system pressure, each pump is equipped with a built-in safety valve and/or a combination pressure control valve. When the pressure in the system exceeds, the oil, overcoming the force of the spring, opens the safety valve and again enters the pump inlet, which helps protect both the pump and other elements of the system from excessive loads. High-capacity pumps are equipped with pressure regulators, an example of which is shown in Fig. 6.3 *c*.

The pressure regulator uses a two-stage regulation system. An element of the first stage is a bypass valve that separates the cavity of the oil system from the inlet to the pump. A throttle insert is installed in the valve housing, through which the oil enters the closed space of the spring chamber

In normal mode, oil passes through both filter chambers, which corresponds to the position of the mark on the spindle of the three-way valve as shown in Fig. 6.5 *b*. When replacing the filter element, the spindle is moved to the position shown in Fig. 6.5 *c*. In this case, the right filter chamber is taken out of operation, the cover can be removed from it and the filter elements can be replaced.

Filter elements (Fig. 6.5 *d*) are made of special paper that is capable of trapping particles larger than 15 μm. The filter element frame is made of metal mesh and stamped covers, which are installed on a perforated sleeve. These elements provide additional protection in case the paper liner fails or bypasses. The housings of the filter chambers are equipped with bypass valves, which open when the pressure drop across the filter element exceeds 0.2...0.3 MPa, which indicates its clogging. In this case, unrefined oil enters the engine. Modern engine filters are equipped with visual indicators and/or electrical switches connected to an automatic alarm system to warn of high filter pressure drop.

In recent years, automatic self-cleaning filters have begun to be used on marine engines as a stand-alone unit or in addition to the traditional filter elements discussed above. An example of such a unit produced by Boll Filters is shown in Fig. 6.6.

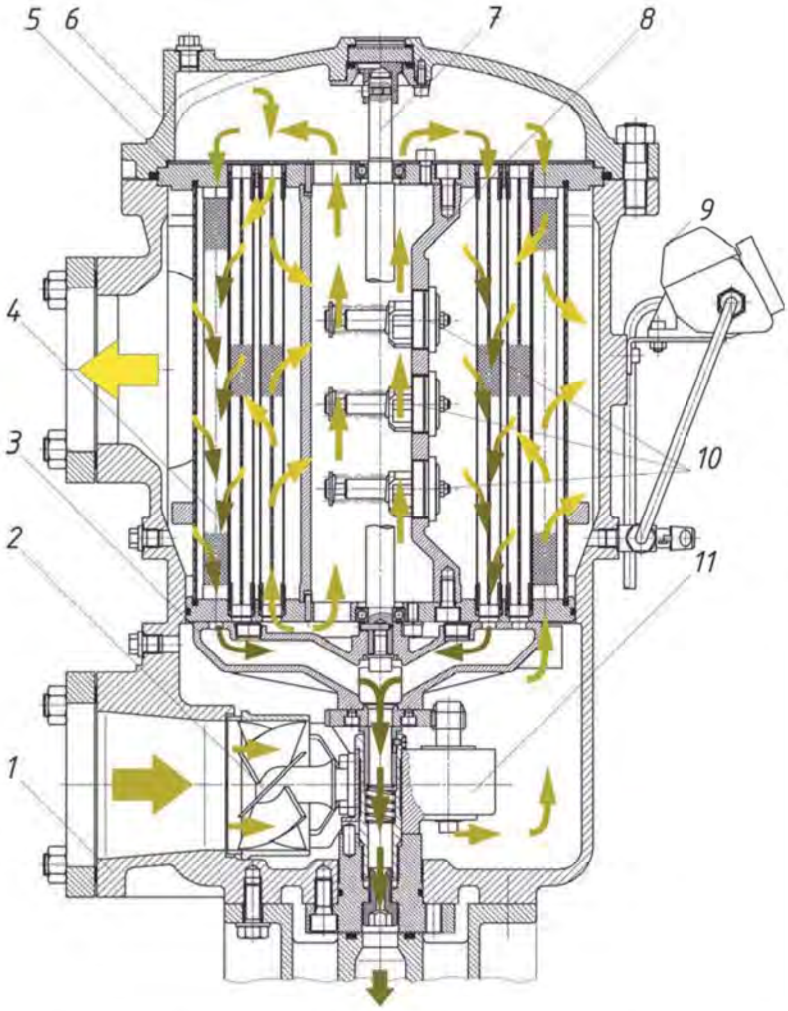


Figure 6.6 – Automatic self-cleaning filter series 6.46 from Boll Filters: 1 – filter housing; 2 – drive impeller; 3 – sludge collection support; 4 – filter elements; 5 – top board; 6 – filter cover; 7 – axis of rotation of the caliper; 8 – diaphragm; 9 – control panel; 10 – bypass valves; 11 – caliper drive gearbox (adapted from [8])

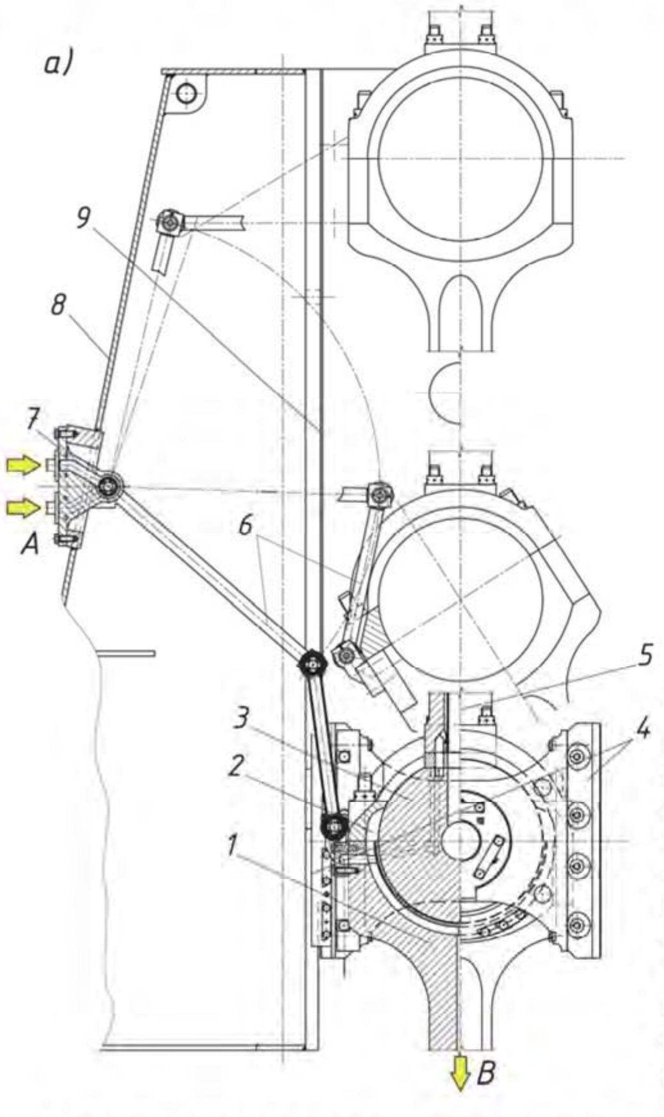
The filter consists of a housing in which two metal boards with holes for installing slot-type filter elements are clamped at the top and bottom. Each element is a metal sleeve made of perforated metal, on the surface of which a metal mesh with a special weave is wound, capable of retaining particles larger than 25 μm. Due to their elongated cylindrical shape, these filter elements are called filter candles.

During operation of the filter, oil passes through the inlet flange, in which a hydraulic turbine wheel is installed to drive the flushing device. After the turbine, part of the unrefined oil flow enters

In this case, the low and medium pressure oil systemsare connected to each other through a check valve. If the medium pressure oil pump fails, the crosshead bearing oil system is supplied with oil pressure from the main bearing. Under such conditions, the engine can only be operated at reduced load.

Crankshafts of low-speed diesel engines usually do not have drillings for oil supply. Lubricant to the frame bearings is supplied from the low-pressure oil pipeline passing inside the diesel frame. Oil flows to the crank bearings through a drilling in the connecting rod from the crosshead assembly.

To supply oil to the moving crosshead assembly of low-speed engines, telescopic pipes (MAN and Mitsubishi) or toggle mechanisms (Sulzer, Wärtsilä, WinGD) are used. The general structure of the toggle mechanism, as well as the order of its operation, are shown in Fig. 6.10 *a*. The telescopic mechanism for supplying oil to the crosshead unit is shown in Fig. 6.10 *b*.



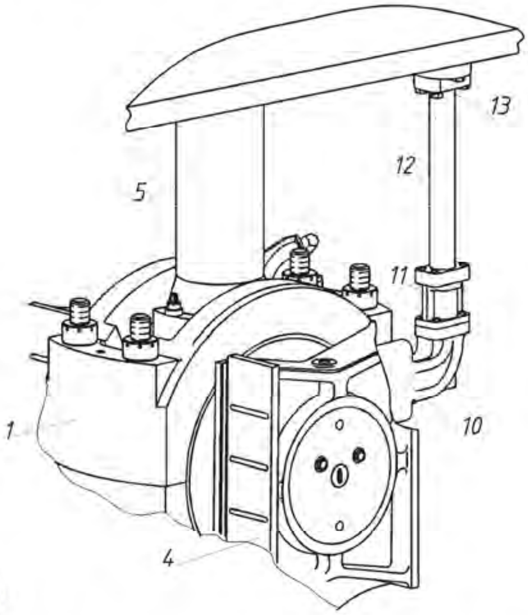


Figure 6.10 – Knee-lever mechanism for supplying oil to the crosshead unit of low-speed Sulzer engines (*a*) and telescopic mechanism for MAN engines (*b*): 1 – connecting rod; 2 – oil distribution groove; 3 – crosshead crossmember; 4 – sliders; 5 – piston rod; 6 – hollow levers of the oil supply mechanism; 7 – oil supply unit to the crosshead; 8 – engine crankcase; 9 – parallels; 10 – connecting unit; 11 – flanges for fastening the telescopic pipe; 12 – telescopic pipe; 13 – oil seal; A – oil rootstock from the medium pressure circuit; B – oil supply for lubrication of the lower connecting rod bearing (adapted from [11, 12])

For engines with timing valves driven by a camshaft, oil is supplied to their hydraulic drive system from the medium pressure oil system through an air separator. Further, at the moment the valves open, the actuators raise the oil pressure in the system to approximately 16 MPa. Other hydraulically driven engine mechanisms are typically driven by oil from a medium pressure system.

6.3 Cylinder lubrication systems

Lubricating lubrication systemsused on powerful two-stroke and some four-stroke engines are open-loop, that is, oil is supplied once to the friction surface of the working cylinder, where it is ir-

to the surface of the cylinder mirror. In the system under consideration, each valve is equipped with a spring pressure accumulator (Fig. 6.12 *f*), the purpose and operating principle of which will be discussed below.

In addition to systemswith mechanical drive of pumping elements, manufacturers of marine engines and components for them have developed pump modules with hydraulic drive and electronic flow control. Such systemsare best suited for integration into electronic engine control systemsand at the same time suitable for use in traditional mechanically controlled engines. The general arrangement of the elements of such a system developed by SKF Lubrication SystemsGermany GmbH and designated CLU4-C is shown in Fig. 6.13. The main element of such a system is the hydraulic lubricating oil supply module, one of the options is shown in Fig. 6.14. The goal of developing CLU 4 was to bring oil consumption into line with the main engine operating modes and operating conditions. The main factors that are taken into account during system operation include: engine speed, load, technical condition, quality and composition of fuel and lubricating oil.

As can be seen in Fig. 6.13, the main elements of the CLU 4 system are;

- hydraulic oil supply modules with electronic control, each of which serves a separate engine cylinder and is designed to connect from 3 to 12 oil valves;
- oil valves installed on the cylinder liner;
- a pump unit supplying oil to the system;
- control units, both individual modules and the entire system as a whole;
- a reservoir filter with a scale for measuring the total oil consumption.

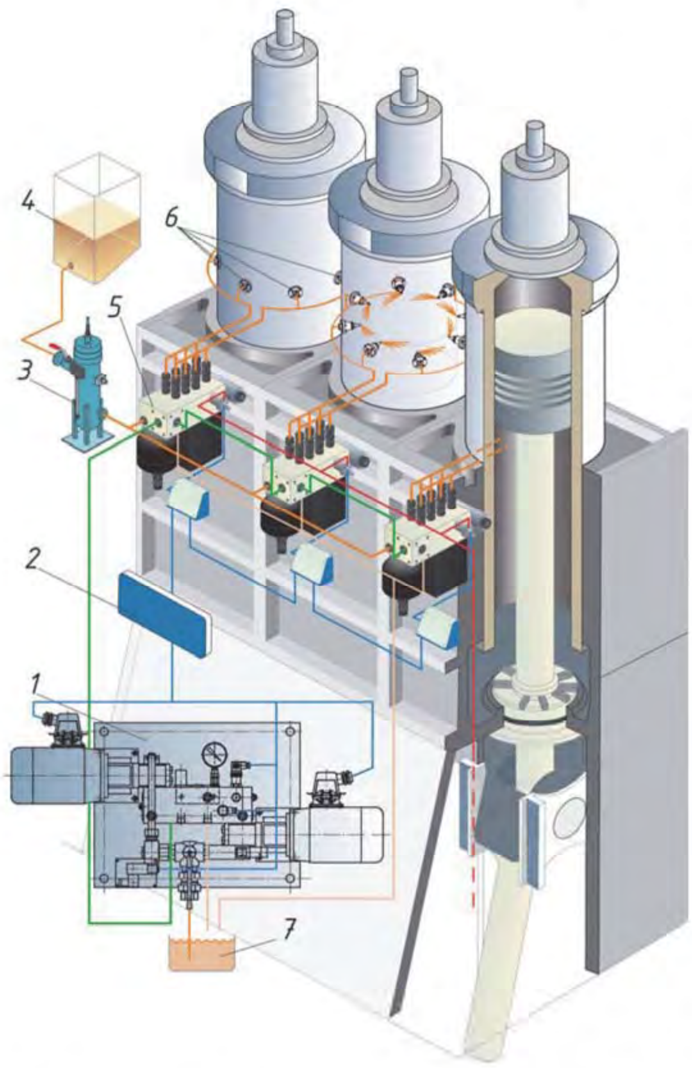


Figure 6.13 – Main elements of the working cylinder lubrication system: 1 – control oil supply module; 2 – electronic control unit; 3 – cylinder oil filter; 4 – cylinder oil supply tank; 5 – hydraulic cylinder oil supply unit; 6 – nozzles for supplying oil to the working cylinder; 7 – tank for hydraulic oil (adapted from [15])

Figure 6.14 – General view of the hydraulic cylinder oil supply unit (adapted from [16])

Similar systemshave been developed by other marine engine manufacturers. For example, the working cylinder lubrication system developed by MAN for mechanically controlled engines,

along the valve tip and all the oil enters the gaps between the rings and the piston.

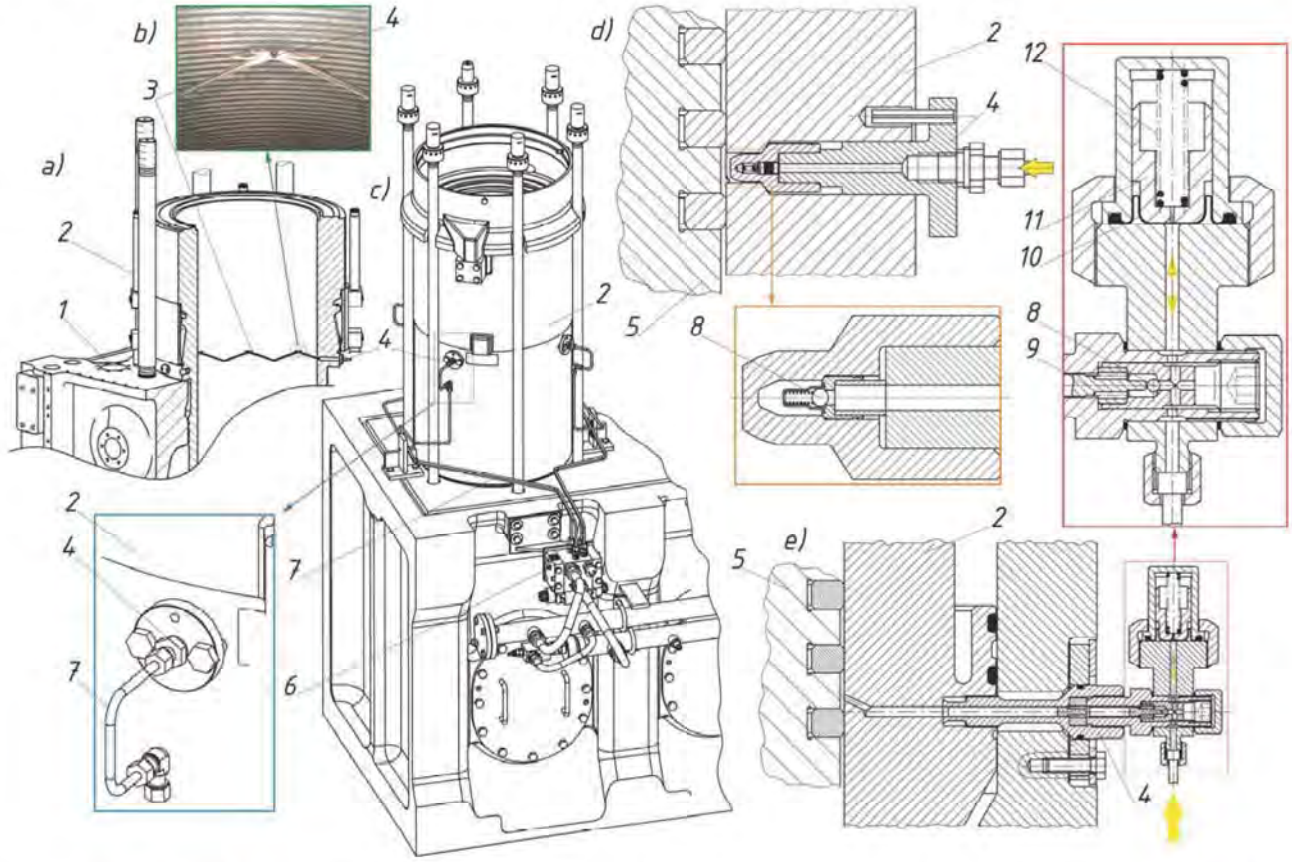


Figure 6.17 – Location of the main elements of the lubrication system for working cylinders and the main types of valves for supplying oil to the working surface of the cylinder liner: *a* – location of the oil distribution grooves on the working surface of the cylinder liner; *b* – general view of the oil distribution grooves; *c* – location of the hydraulic oil supply module, oil lines and oil supply valves on the X-40 engine from WinGD; *d* – oil supply valve for the X-40 engine from WinGD; *e* – oil supply valve with spring accumulator for Sulzer RTA series engines; 1 – cylinder block; 2 – cylinder liner; 3 – oil distribution grooves; 4 – oil supply valve; 5 – piston; 6 – hydraulic module for supplying cylinder oil; 7 – oil lines; 8 – ball valve; 9 – rod; 10 – separating diaphragm; 11 – accumulator piston; 12 – accumulator spring (adapted from [11, 19])

To prevent pulsed oil supply, some manufacturers equip each oil supply valve with a piston pressure accumulator. The design of such a valve is shown in Fig. 6.17 *f*. The oil supplied by the pump section enters the valve cavity, from where it enters the gap between the internal channel and the rod placed in it through the ball valve. Moving along the gap, the flow speed slows down, and oil enters the cylinder at a speed insufficient to separate it from the nozzle hole. At moments when the oil injection rate exceeds the capacity of the oil outlet channel for cylinder lubrication, excess oil enters the spring accumulator, overcoming the spring force, lifting the piston and deforming the separating diaphragm. This way the accumulator is charged. After the oil supply from the discharge element stops, the accumulator gradually displaces the accumulated oil through the nozzle hole onto the surface of the cylinder liner. Naturally, the feeding process is only possible when the oil pressure in the valve cavity is higher than the pressure in the working cylinder and the rod does not hold the ball valve closed. Thus, oil supply begins automatically as soon as the pressure at the nozzle exit is lower than the lubricating oil pressure, for example, when passing a set of piston rings.

A fundamentally different approach in the field of lubrication of working cylinders is to supply oil to the working surface of the bushing under high pressure in the form of a finely sprayed oil aerosol (Fig. 6.18 *a*).

This technology was developed by Hans Jensen Lubricators and was called SIP, which is an ab-

the overflow tube, the tank continues to fill. The air in its upper part begins to compress, the pressure in the tank increases, and the drainage speed increases. This continues until a balance is established between the inflow and outflow of oil from the reserve tank. In this case, a certain excess pressure is established in the system, which is constantly maintained during engine operation.

When the engine stops, the pressure in the lube oil system decreases. In this case, the check valve at the inlet to the engine oil system closes, and the check valve on the reserve tank opens. The oil, displaced by compressed air, begins to flow under pressure from the emergency lubrication system back into the turbocharger to the bearing bushings. When the lubricating oil level drops below the overflow tube (level 2), the excess pressure in the system ceases, but oil continues to flow to the turbocharger bearings by gravity at a reduced flow rate. As soon as the height of the supply (drain) pipe is reached (level 1), the remaining volume in the tank flows out through a small calibrated hole in its lower part. This ensures three-stage lubrication of the turbocharger bearings during the rundown of its rotor. The most intense pressure is at the moment the engine stops, gravitational pressure is of medium intensity when the speed has dropped significantly and low intensity when the rotor has practically stopped.

6.5 Friction and lubrication modes

In friction pairs between individual engine structural elements, various friction and lubrication modes can occur depending on the speed and method of relative movement of the rubbing surfaces, the loads acting on them, the amount and method of oil supply. There are three main modes of friction (lubrication):

- hydrodynamic;
- contact-hydrodynamic;
- borderline.

Hydrodynamic or liquid mode is a lubrication method in which the thickness of the oil layer is sufficient to prevent direct contact of micro-irregularities of the rubbing surfaces. If the surfaces are separated by a relatively thick layer of oil (more than 0.1 μm), then the properties of the oil in the film and in the volume are the same.

In the absence of rotation, the shaft, under the influence of loads and its own weight, occupies an eccentric position, and on both sides of it a wedge-shaped gap is formed with a maximum gap in the upper part Δ (Fig. 6.21 *a*).

When the shaft rotates, the layer connected to its surface by sorption forces carries along subsequent layers and forces them into the narrowed part of the wedge-shaped gap. The mutual connection between the film adsorbed on the surface of the shaft and the oil in the gap is due to viscous forces (Fig. 6.21 *b*). Thus, the oil is involved in a relative movement in the direction of rotation and in the longitudinal direction towards the ends of the bearing. The limited clearances prevent the free flow of oil, which is practically incompressible under these conditions. Therefore, hydrodynamic pressure is created in the gap, acting on the shaft, which increases in the direction of decreasing the size of the slit along the circumference. In the longitudinal direction, the pressure diagram has the form of a hyperbola (Fig. 6.21 *c*). As a result, the shaft is raised and shifted in the direction of rotation (Fig. 6.21 *b*) by an amount e. Some of the oil flows out through the ends of the bearing, and the rest is pumped through the narrow part of the gap. In this case, the shaft is raised by an amount $h_{\min}$ until an equilibrium state occurs, at which the flow area will be sufficient to pass the remaining part of the oil.

The hydrodynamic pressure developed in the lubricant layer ensures the separation of the shaft and bearing surfaces, and the friction between the surfaces is replaced by internal friction between the oil layers.

As the rotation speed increases, the hydrodynamic pressure and the thickness of the oil film increase, the shaft floats up, and the friction force quickly decreases. With a further increase in rotation speed, the bearing capacity of the oil layer increases, but at the same time the forces of liquid friction in the layer increase.

SECTION 7

Heat flows in marine engines. Heat removal systems from their structural elements

7.1 Thermal balance of marine engines

In internal combustion engines, only part of the heat in the process of converting thermal energy is converted into mechanical energy. For modern engines, this figure ranges from 38...55%; the rest of the heat is transferred to the environment in one way or another and represents irreversible heat loss. Part of the heat with exhaust gases is dissipated in the atmosphere, and part, by heat transfer through engine elements, is transferred to the cooling system and then to the sea water. A certain portion of the heat undergoes complex transformations, for example, part of the mechanical work is spent on overcoming frictional forces with subsequent conversion into thermal energy, which is also discharged into the cooling system. Some of the heat is returned to the cycle with air supplied from the turbocharger. The distribution of the heat released during fuel combustion in the engine Q_f into useful heat and various types of heat losses is called *external heat balance.* In Fig. 7.1 shows a diagram of the distribution of heat flows resulting from fuel combustion in a modern combined engine.

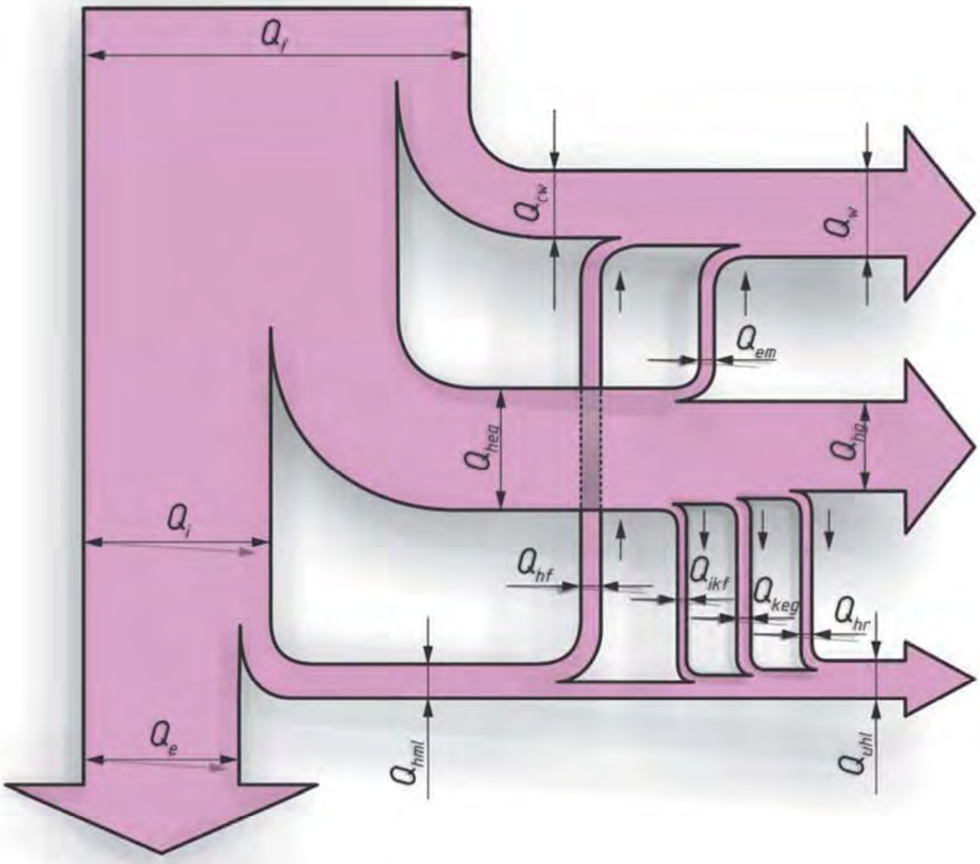


Figure 7.1 – Engine thermal balance diagram: Q_f – heat of combustion of fuel; Q_i – heat spent on indicator work; Q_e – heat spent on efficient work; Q_w – heat removal with coolant; Q_{cw} – heat loss due to heat exchange with the cylinder walls; Q_{hf} – heat loss from friction; Q_{em} – heat loss to cooling or the environment from the exhaust manifold; Q_{heg} – heat loss with gases after the cylinder; Q_{hg} – heat removed with exhaust gases; Q_{uhl} – total unaccounted heat losses; Q_{hml} – heat equivalent to mechanical losses; Q_{ikf} – heat loss from incomplete combustion of fuel; Q_{keg} – heat equivalent to the kinetic energy of gases; Q_{hr} – heat loss with radiation

ues of heat flows, kW/m^2:

$$q_{w1} = \alpha_g (T_g - T_1);$$

$$q_{w2} = \frac{\lambda_p}{\delta_p}(T_1 - T_2);$$

$$q_{w3} = \alpha_w (T_2 - T_w);$$

where α_g, α_w are the heat transfer coefficients of gas and cooling water; T_1, T_2, T_w – wall temperatures, respectively, on the gas side, on the cooling side and on the cooling water side.

The ratio λ_p/δ_p is determined as:

$$\frac{\lambda_p}{\delta_p} = \sum_{i=1}^{n} \frac{\lambda_i}{\delta_i},$$

where i is the number of layers of a conditional wall – soot, main layer (metal), scale; λ_i, δ_i – heat transfer coefficients and thickness of these layers.

Total temperature difference during heat transfer from gases to coolant (water or oil):

$$T_g - T_w = q_{w\Sigma}\left(\frac{1}{\alpha_g} + \frac{\delta_p}{\lambda_p} + \frac{1}{\alpha_w}\right),$$

where is the total heat flow:

$$q_{w\Sigma} = \frac{T_g - T_w}{\frac{1}{\alpha_g} + \frac{\delta_p}{\lambda_p} + \frac{1}{\alpha_w}} = k(T_g - T_w),$$

where $1/\alpha_g$ is the thermal resistance to heat transfer from the gas to the wall; δ_p/λ_p – thermal resistance to thermal conductivity through the wall; $1/\alpha_w$ – thermal resistance to heat transfer from the wall to the coolant.

The overall heat transfer coefficient from gases to the coolant in general terms, kW/(m^2×K):

$$k = \frac{1}{\frac{1}{\alpha_g} + \frac{\delta_p}{\lambda_p} + \frac{1}{\alpha_w}}.$$

In a stationary heat exchange process, temperatures T_1 and T_2 are taken as steady. In the actual process, on the surface layer of the main wall, due to the variable nature of the heat supply and loss in the soot layer, the temperature is somewhat different from that accepted as the calculated T_1, including due to temperature fluctuations. These fluctuations (3...5% of T_1) along the thickness δ_p are damped and the temperature at a certain thickness (1...1.5 mm from the interface) practically stabilizes. This is explained by the high thermal inertia of the metal, due to which the temperature of the parts does not follow the change in the temperature of the gases in the cylinder. Small temperature fluctuations are observed only on the very surface of the walls of the combustion chamber.

7.3 Temperature fields of parts of the cylinder-piston group

The main indicators of the thermal state of parts of the cylinder-piston group are temperatures in the most critical places and in places where they interface with other parts (Fig. 7.5). The permissible temperature level at each point depends on the operating conditions of the parts. So, for a friction pair: «ring-cylinder bushing», the maximum permissible temperature is determined by the lubrication conditions. Where there is no friction pair, higher temperatures may be acceptable. The thermal state of structural elements, along with other factors, such as mechanical stresses from gas pressure, tightening forces of anchor ties and fastening studs, significantly affect the reliability of the engine and the service life of its main elements.

the seat (Fig. 7.8). As shown In Fig. 7.8, the hottest area is the center of the valve end with a temperature of 630 °C. The area near the valve seat is much cooler, with a temperature of about 390 °C. Strong temperature gradients lead to increased mechanical stress due to thermal stress in this area.

Operation at high temperatures and high engine load reduces the resistance of valve materials to high temperature corrosion and shortens their service life.

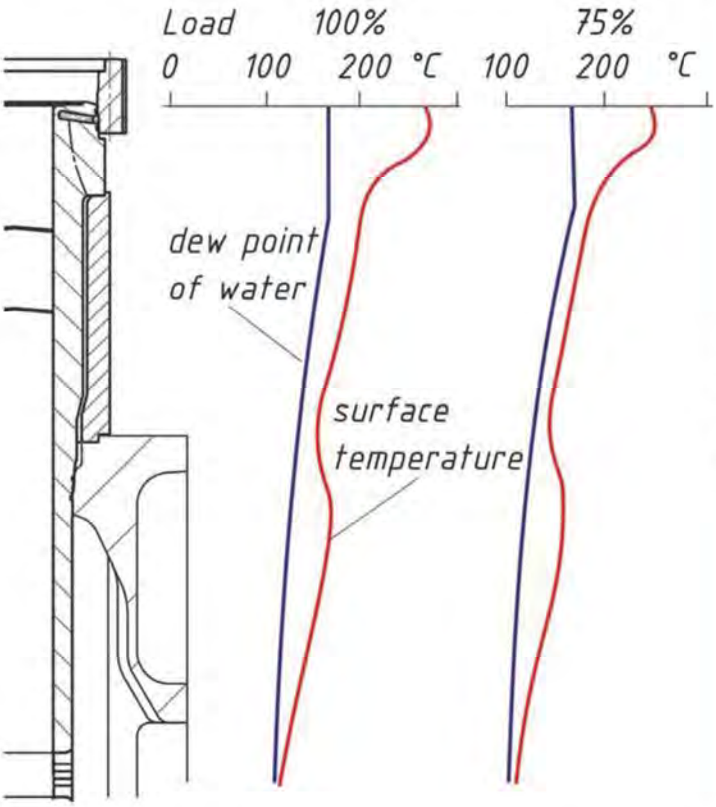


Figure 7.7 – Character of changes in condensation and working surface temperatures along the height of the cylinder liner of a low-speed two-stroke diesel engine (adapted from [5])

The effect of high temperature corrosion in engines operating on heavy fuel oil occurs due to the large amount of vanadium and sodium contained in the heavy fuel oil (up to 600 and 200 min^{-1}, respectively). Vanadium in fuel is contained in the form of fuel -soluble complex organometallic compounds, and sodium is mostly in the form of *NaCl* crystals or an aqueous solution of its salts. During combustion, these elements form various compounds called sodium vanadyl vanadates. In structure, these are semi-liquid salts with a low melting point and high adhesive ability.

The most active compounds of this group are $Na_2O{\cdot}6V_2O_5$ and $5Na_2O{\cdot}12V_2O_5$. During their solidification on the metal surface in the temperature range of 530...600 °C, atomic oxygen is released which reacts with the metal and forms oxides that diffuse into the overlying layer of deposits. As the deposits remelt, oxygen is absorbed from the exhaust gases. Thus, sodium vanadyl vanadates act as an oxygen carrier from exhaust gases to the metal surface, causing intense corrosion.

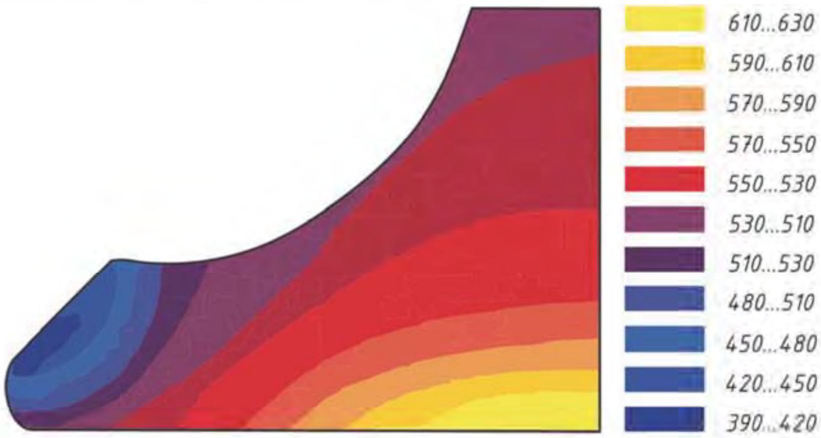


Figure 7.8 – Typical temperature distribution (°C) for the exhaust valve of a modern diesel engine made of Nimonic alloy (adapted from [6])

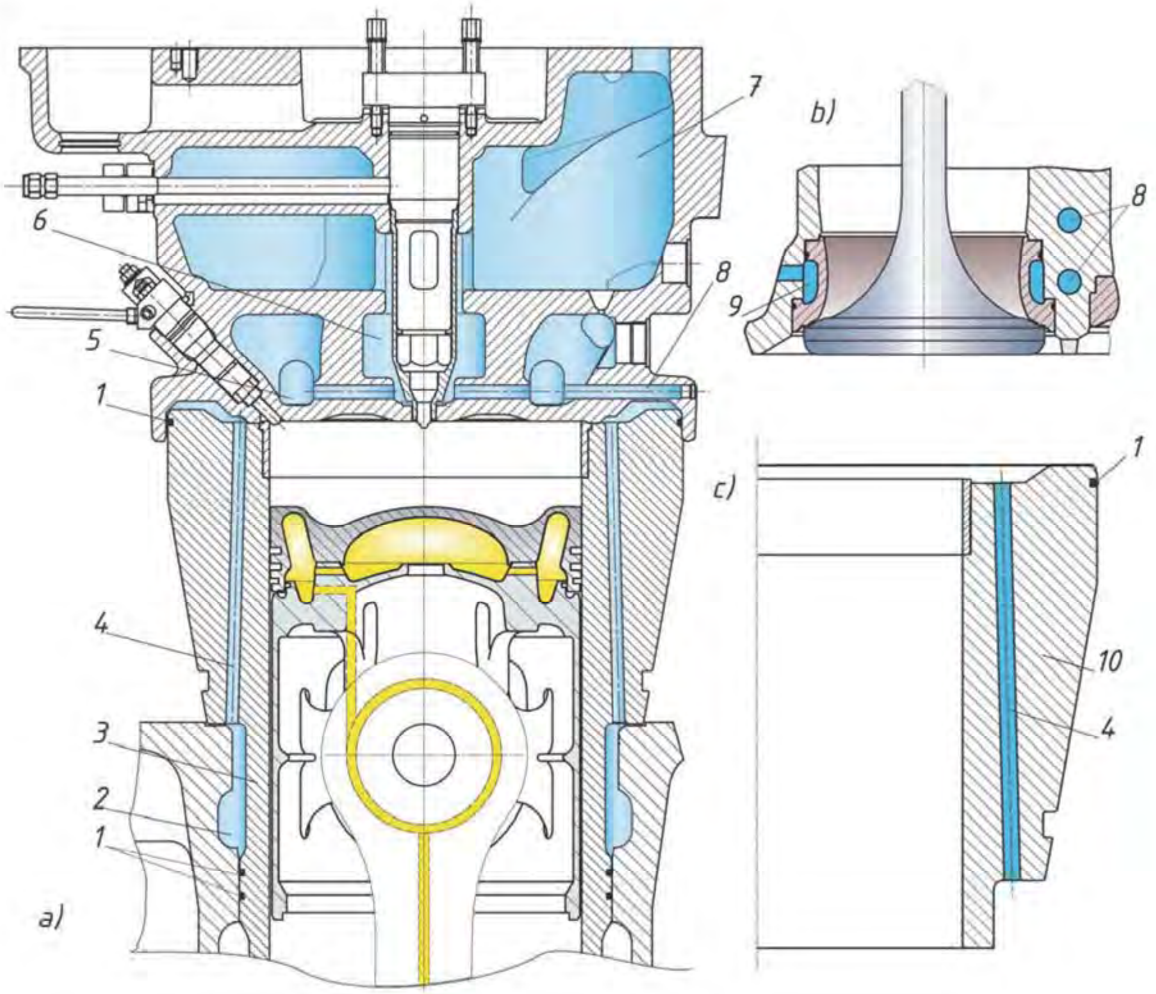


Figure 7.10 – Cooling of the main elements of a four-stroke medium-speed engine 46 series from Wärtsilä. *a* – cooling of the elements forming the engine working space; *b* – cooling of exhaust valve seats; *c* – channel cooling of the firing surface of the cylinder liner; 1 – sealing rings; 2 – cooling jacket; 3 – cylinder liner; 4 – channels for cooling the firing surface of the cylinder liner; 5 – annular water manifold for supplying water to the fire bottom of the cylinder cover; 6 – fuel injector cooling cavity; 7 – cooling cavity of the second tier; 8 – radial drillings for supplying cooling water to the fire bottom of the cylinder cover; 9 – cavity for supplying water for cooling the exhaust valve seat; 10 – upper collar of the cylinder liner (adapted from [9])

In two-stroke engines, water from oblique-radial drillings along the fire bottom of the cylinder cover enters the annular manifold of the formations between the cover and the exhaust valve assembly (Fig. 7.11 *b*). From here, part of the water is supplied to cool the valve seat, and the remaining flow is supplied to cool the valve assembly.

In some cases, on low-speed engines, sea water is used to cool the charge air, which is supplied to the cooler by a separate pump.

For high-performance, medium-speed engines, the charge air cooler is divided into high-temperature and low-temperature sections to achieve maximum heat recovery. By adjusting the intensity of water flows in both sections, stable temperatures of the charge air at the engine inlet are achieved, which is especially important at low and medium loads, when the air needs to be heated.

7.5.2 Piston cooling

To cool the pistons in modern engines, circulating oil is used, which greatly simplifies the system for supplying the cooling medium to the moving pistons and, most importantly, minimizes the consequences of leaks in the event of damage to the seals.

Low-speed MAN engines use circulation cooling of the pistons (Fig. 7.12 *a*), in which oil enters

cools it. The oil supply line to the injectors is separated from the main circulation system by a pressure reducing valve configured in such a way that the oil supply begins only when the pressure in the system rises above a certain value necessary to ensure reliable lubrication of the bearings. As a flow interrupter, a system of drillings and grooves is usually used in the camshaft journals and their bearings, which act as a rotating spool pair.

7.6 Charge air cooling

In modern highly accelerated engines, the charge air cooling system plays a key role in organizing the engine's operating process. Despite the name, in modern engines, the function of the system is not only to cool the charge air when the engine is operating at loads close to maximum, but also to heat it when the engine is operating at low and medium loads. For this purpose, engines are equipped with a two-section cooler, one section of which is included in the HT circuit, and the other in the LT circuit (Fig. 7.15). Thus, the air is first cooled (or heated) in the HT circuit section, and then enters the LT circuit section in which the coolant temperature is regulated by a special thermostatic valve. The valve either connects the LT section to the low temperature circuit if air cooling is required, or disconnects the section if there is no need for cooling. The joint operation of the two sections makes it possible to obtain stable air temperatures at the engine inlet, regardless of its operating mode.

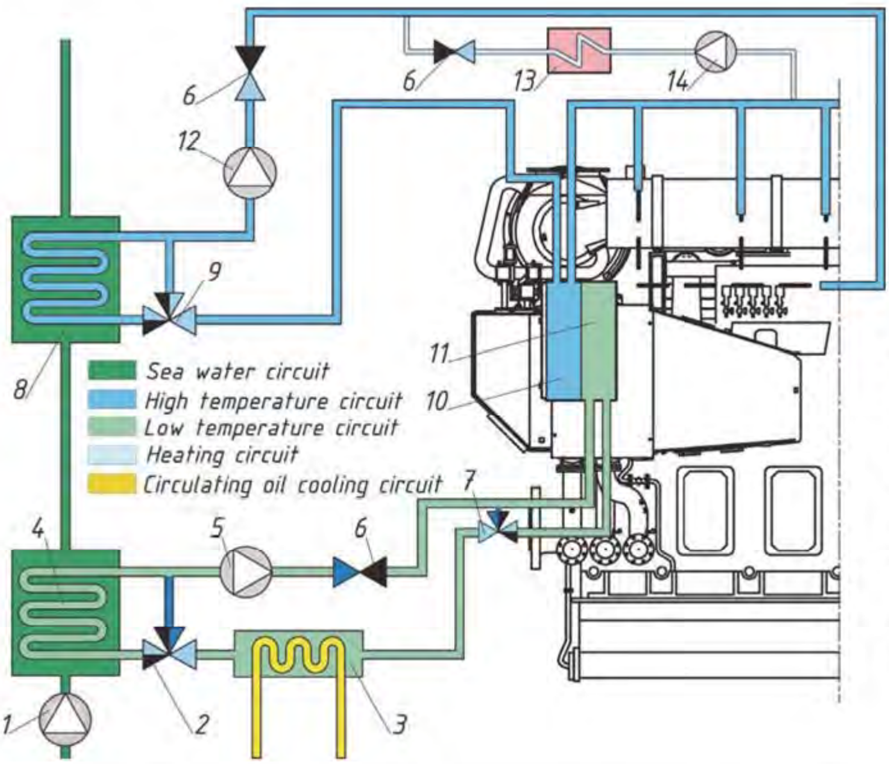


Figure 7.15 – Diagram of connecting the air cooler to the HT and LT circuits of a medium-speed four-stroke engine: 1 – sea water pump; 2 – thermostatic valve LT; 3 – oil cooler; 4 – LT circuit cooler; 5 – LT circuit pump; 6 – shut-off valve; 7 – thermostat of the LT stage of the air cooler; 8 – HT circuit cooler; 9 – thermostatic valve HT; 10 – section LT of the air cooler; 11 – LT section of the air cooler; 12 – – HT circuit pump; 13 – heater; 14 – heating circuit pump (adapted from [14])

Some engines use a special thermostatic valve to control the air temperature at the cooler inlet, allowing water from the high-temperature circuit to pass through the low-temperature section of the air cooler. The valve is controlled by charge air pressure, which directly depends on the operating mode of the engine. An example of such a valve used on engines of the TBD 645 series from DEUTZ MWM is shown in Fig. 7.16.

7.7 Main units and elements of cooling systems

Pumps. Centrifugal pumps are used for the cooling system of diesel engines. They have a fairly high efficiency and are simple in design. The use of centrifugal pumps allows for continuous flow, without pulsation, which ensures uniform cooling of the engine. To protect against the aggressive

lines. Disconnect individual elements from the cooling system so that the rest can function normally. For example, for dismantling cylinder covers, special valves are provided that allow you to disconnect this cover from the system so that you do not have to drain water from the entire system.

The general purpose of the fittings is to turn on or off flows, regulate flow, temperature or pressure of the flow and protect against off-design modes.

According to their purpose, valves are distinguished: shut-off valves (turning on and off flow), control valves (changing or maintaining a given flow rate, pressure, temperature), safety valves (preventing excessive pressure increases, preventing changes in flow direction) and control valves (level indicators). The listed fittings can be installed both on pipelines and on individual units of the cooling system.

According to the control method, a distinction is made between valves with manual, electric, hydraulic, pneumatic drives and self-acting ones, including pulse ones, driven by the medium itself. The most widespread in cooling systems are shut-off valves, valves and taps of various designs (Fig. 7.19), as well as check and safety valves.

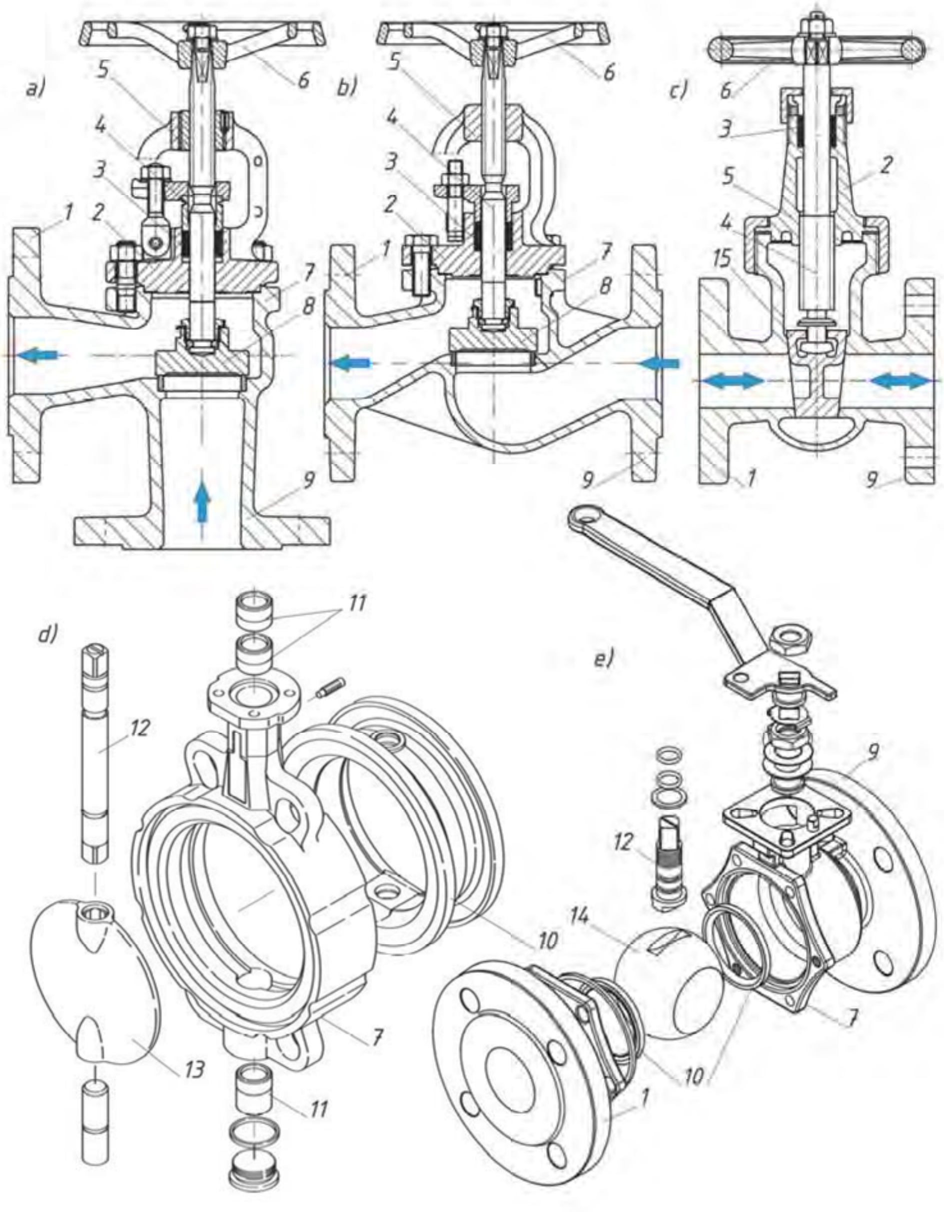


Figure 7.19 – Shut-off devices used in cooling systems of marine engines: *a* – angle non-return shut-off valve; *b* – horizontal non-return shut-off valve; *c* – clinker valve; *d* – butterfly type rotary valve; *e* – ball valve; 1 – output flange; 2 – cover; 3 – stuffing box seal; 4 – lead screw (spindle); 5 – running nut; 6 – flywheel; 7 – housing; 8 – locking plate; 9 – inlet flange; 10 – polymer seals; 11 – shaft bushings; 12 – spindle; 13 – damper; 14 – ball locking element; 15 – wedge valve (adapted from [20, 21, 22])

SECTION 8

Starting and reversing systems

8.1 Engine starting process

A feature of the operating process of piston engines is that the working stroke in the cylinder must necessarily be preceded by the compression stroke, and in diesel engines the charge temperature at the end of compression must be sufficient for self-ignition of the supplied fuel. To fulfill these two conditions, when starting the engine, it must first be spun up using an external energy source. This is the function that the engine starting system performs. If the engine is reversible, the starting system must ensure that the engine starts in both directions, in which case it is called a start and reverse system.

8.2 Electric starting

On small engines that are used as emergency diesel generators or to drive lifeboats, starting with electric starters is widely used (Fig. 8.1).

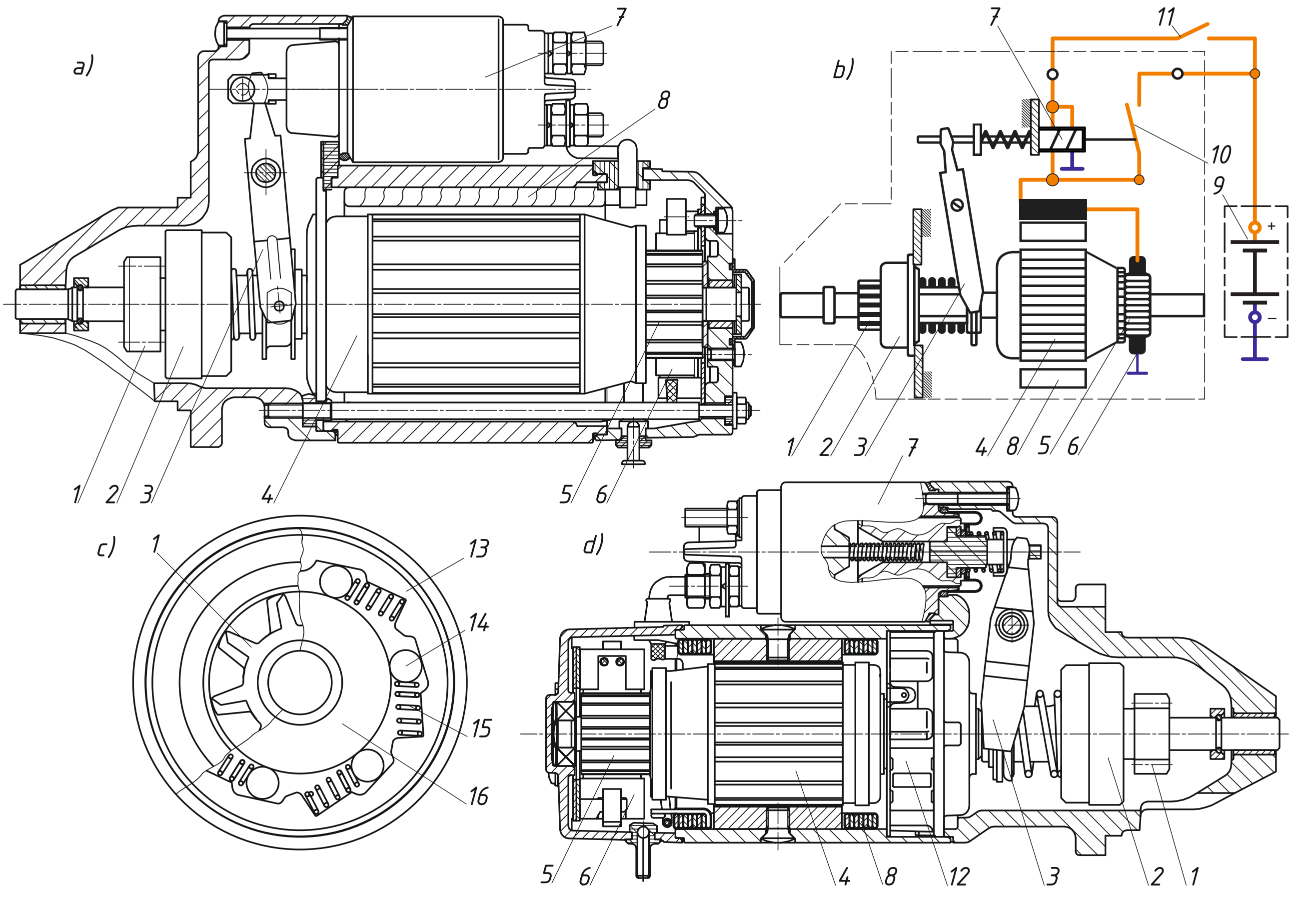


Figure 8.1 – Electric starter with direct drive (*a*) and its connection diagram (*b*), roller safety clutch (*c*), electric starter with reduction gear (*d*): 1 – engagement gear with ring gear on the flywheel; 2 – roller safety clutch; 3 – drive lever for engaging the gear; 4 – rotor; 5 – collector; 6 – brush mechanism; 7 – solenoid relay; 8 – stator winding; 9 – battery; 10 – central switch of the retractor relay; 11 – starter switch; 12 – planetary gearbox; 13 – safety clutch stator; 14 – rollers; 15 – spring; 16 – safety clutch rotor (adapted from [4])

The electric starter is a DC commutator engine, with direct drive (Fig. 8.1 *a*) or driven through a reduction gear (Fig. 8.1 *d*), powered by a starter battery (Fig. 8.1 *b*). Starter engines have a special design with four brushes, which increases the armature current density and starter power.

When the starter is turned on, electric current (through the start relay) is supplied to the solenoid coil of the solenoid relay. The solenoid core retracts and, through a linkage, engages the starter engine gear with the ring gear on the engine flywheel. After this, the contacts of the starter relay close, connecting it to the battery and switching power to the holding winding of the solenoid. The relay contact group is designed for high starting current, reaching hundreds of amperes. To protect the electric motor from excessive spinning after starting the engine, the starter gear is connected to the shaft through a ratchet or roller-type overrunning clutch (Fig. 8.1 *c*), which ensures one-way transmission of mechanical energy from the starter to the flywheel. If the flywheel speed increases above the starting speed, the clutch wedges and ensures free rotation of the gear without transmitting torque to the electric motor shaft. After the starter is turned off, the drive gear returns to its original position.

Electric starting is relatively simple and is especially convenient when remote controlled, but starter batteries are heavy, difficult to maintain, have a relatively short service life and high cost.

8.3 Pneumatic systems for starting and reversing engines

On marine engines, compressed air energy is used as the main type of starting energy.

There are two main ways to air start engines.

Direct launch. Compressed air is supplied directly to the working cylinders of the engine in accordance with their operating order. Air pressure forces the pistons to move from TDC to BDC, which causes the crankshaft to turn. This starting method has no power limitations and is therefore widely used on both low-speed and large medium-speed engines.

Starting with a pneumatic starter. Compressed air is used to drive a pneumatic engine, which, at the time of start-up, is engaged with the ring gear on the flywheel, driving the diesel crankshaft. This starting method is used primarily for engines of medium and low power.

8.3.1 Pneumatic starters

Pneumatic starters are distinguished by a wide variety of designs, both in layout schemes and in methods of converting compressed air energy into mechanical work. They have a number of advantages over other starting methods, which in many cases make their use preferable for economic and technical reasons.

These benefits include:

- simplicity of the starting system, which does not require intervention in the design of the engine itself;
- ease of regulation of rotation speed and torque;
- the possibility of full braking under load without compromising the design and performance characteristics of the pneumatic engine;
- no overheating;
- large resource of work;
- complete fire and explosion safety;
- insensitivity to adverse environmental factors (dust, moisture, etc.).

Conventionally, all existing designs of pneumatic starters can be divided into two main types: with positive-displacement or dynamic-action engines.

8.3.1.1 Pneumatic starters with positive displacement engines

Gear pneumatic engines are widely used as positive displacement engines in starters, in which compressed air acts on two gear rotors rotating in a stator tightly fitted to them.

Gear pneumatic engine. In the air starter shown in Fig. 8.2, one of the gear rotors transmits rotation to an internal gearbox. This allows you to increase the torque on the shaft. The second rotor rotates freely on supports installed in the housing. Compressed air supplied to the working chamber acts on the side surfaces of the gear teeth. In this case, forces arise equal to the product of com-

In any circuit, two circuits can be distinguished, the power air circuit and the control circuit. If the power of one starter is insufficient to reliably start the engine, several of them can be installed – from two to three, connected in parallel. This allows not only to increase starting power, but also to increase the reliability of the starting system by redundant units of the same type.

8.4 Direct starting systems

The direct air starting system of a marine internal combustion engine is used to start, stop (in an emergency) or reverse the engine.

The starting system elements are mounted directly on the engine and provide compressed air to the working cylinders in accordance with the in accordance with the order of their work. Air enters each cylinder under a pressure of 3...1.5 MPa at an angular interval of 2...110° after TDC, which corresponds to the expansion stroke. For four-stroke diesel engines, starting from any position of the crankshaft is ensured if there are at least six cylinders; for two-stroke diesel engines, at least four. Starting with compressed air can be done both with simultaneous supply of fuel to the cylinders (mixed start) and without it (separate start). The supply of compressed air on board must provide at least six consecutive starts for non-reversible engines and 12 starts for reversible engines. Particularly difficult starting conditions are created in main marine diesel engines with direct transmission to a fixed-pitch propeller, since the energy of the starting air must overcome not only the forces of resistance to the rotation of the engine itself, but also the resistance to rotation of the propeller with the masses of water attached to it.

Modern engines predominantly use separate starting, which is preceded by slow cranking of the engine, for which a separate circuit is provided in the starting system. Slow cranking is necessary to monitor the condition of the mechanical part of the engine, check that coolant does not enter the cylinders due to their depressurization, etc. Engine automation features include a slow crank mode when starting the engine after stopping for more than 30 minutes. If after several revolutions of the crankshaft no problemsare detected, the start mode is automatically activated.

The startup process can be divided into three stages:

- cranking of the engine in the initial period under the influence of starting air pressure entering the cylinder, the piston of which was in the starting position;
- subsequent acceleration of the engine under air pressure entering the remaining cylinders in accordance with the order of their operation;
- transition of the engine to operation on fuel.

To shut off the fuel supply during engine acceleration on positive displacement fuel pumps of low-speed engines, a bypass valve with a pneumatic drive is provided. On medium-speed engines, at the moment of starting, the fuel pump racks are switched to zero flow. In electronically controlled engines, the supply is switched off by the central control unit.

A schematic diagram of the direct starting system for medium-speed engines 46 series from Wärtsilä is shown in Fig. 8.6.

The air line from the starting air receiver is equipped with a shut-off valve and a check valve, as well as a purge valve installed in front of the main starting air valve.

The main start valve and the slow rotation valve are actuated by control air supplied through solenoid control valves mounted on the local instrument panel by pressing the start button or by activating the solenoids through the remote control system. Slow cranking of two turns is activated automatically if the engine has been stopped for more than 30 minutes. During slow rotation, air enters the slow rotation valve through the regulator, which reduces the pressure to the required level.

A check valve in the slow rotation line prevents air leakage during main startup.

The shut-off valve prevents air leakage when cranking slowly.

After activating the main start valve, the air is divided into two streams, power and control. The first is fed through flame arresters to the starting valve on the cylinder cover. The second air flow is supplied to the starting air distributor, which sequentially supplies control air to the starting valves

spool pairs located radially relative to the axis of rotation of the drive shaft. The distributor spools are controlled by a cam shaft driven by the engine camshaft. In the absence of pressure in the starting system, the spools are held by springs in the upper position, in which the starting valves connected to the spool pairs communicate with the atmosphere (Fig. 8.9 *c*). When the main starting valve opens, the spool pistons are pressed against the cam, causing the control piston for the engine cylinder that is in the starting position to pass control air to the starting valve (Fig. 8.9 *d*). The starting valve opens and supplies compressed air to the engine cylinder, causing the crankshaft to turn. As a result, the distributor cam shaft is also rotated, opening the air supply to the starting valve of the next cylinder. In this case, the previous starting valve is connected to the atmosphere through a spool pair. While the main starting valve is open, the sequential opening of the spools will be repeated until the crankshaft speed rises to the level necessary for self-ignition of the fuel. After the main start valve closes, the pressure drops quickly and springs lift the spools away from the cam. Thus, the spools interact with the cam only during the starting cycle and therefore wear is negligible.

To reverse the engine, the cam shaft is equipped with two profiles – forward or reverse. Profiles are changed by axial movement of the shaft using a pneumatic (hydraulic) drive. A cylinder is used as an actuator, in which a ledge moves, performing the function of a piston dividing the working space of the cylinder into two volumes. The supply of air (oil) to the corresponding cavity leads to axial movement of the cam shaft (Fig. 8.9 *b*).

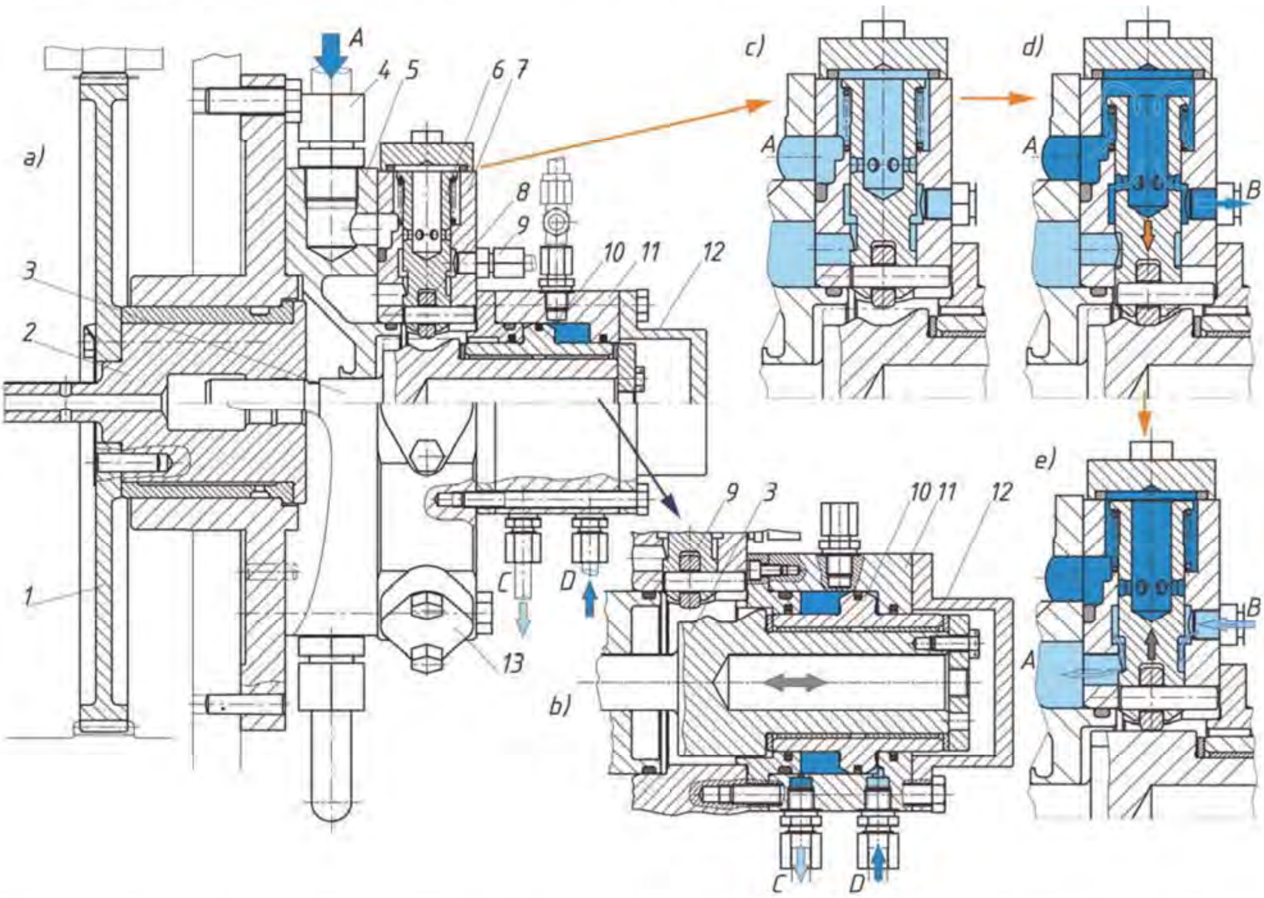


Figure 8.9 – Spool distributor of compressed air across the cylinders of Wärtsilä RTA series engines; general device (*a*), mechanism for changing profiles of distributor cams when reversing the engine (*b*); flow diagram (*c*, *d*, *e*). 1 – drive gear; 2 – hub; 3 – drive shaft with a double set of cams; 4 – air supply fitting from the main starting valve; 5 – distributor housing; 6 – spool; 7 – spool bushing; 8 – roller pusher; 9 – air supply fitting to the starting valves; 10 – piston for axial movement of the shaft; 11 – cylinder for axial movement of the shaft; 12 – cylinder cover; 13 – spool module cover *A* – air supply from the main starting valve; *B* – air to the starting valves of the working cylinders; *C*, *D* – air to the piston drive for axial movement of the shaft (adapted from [11])

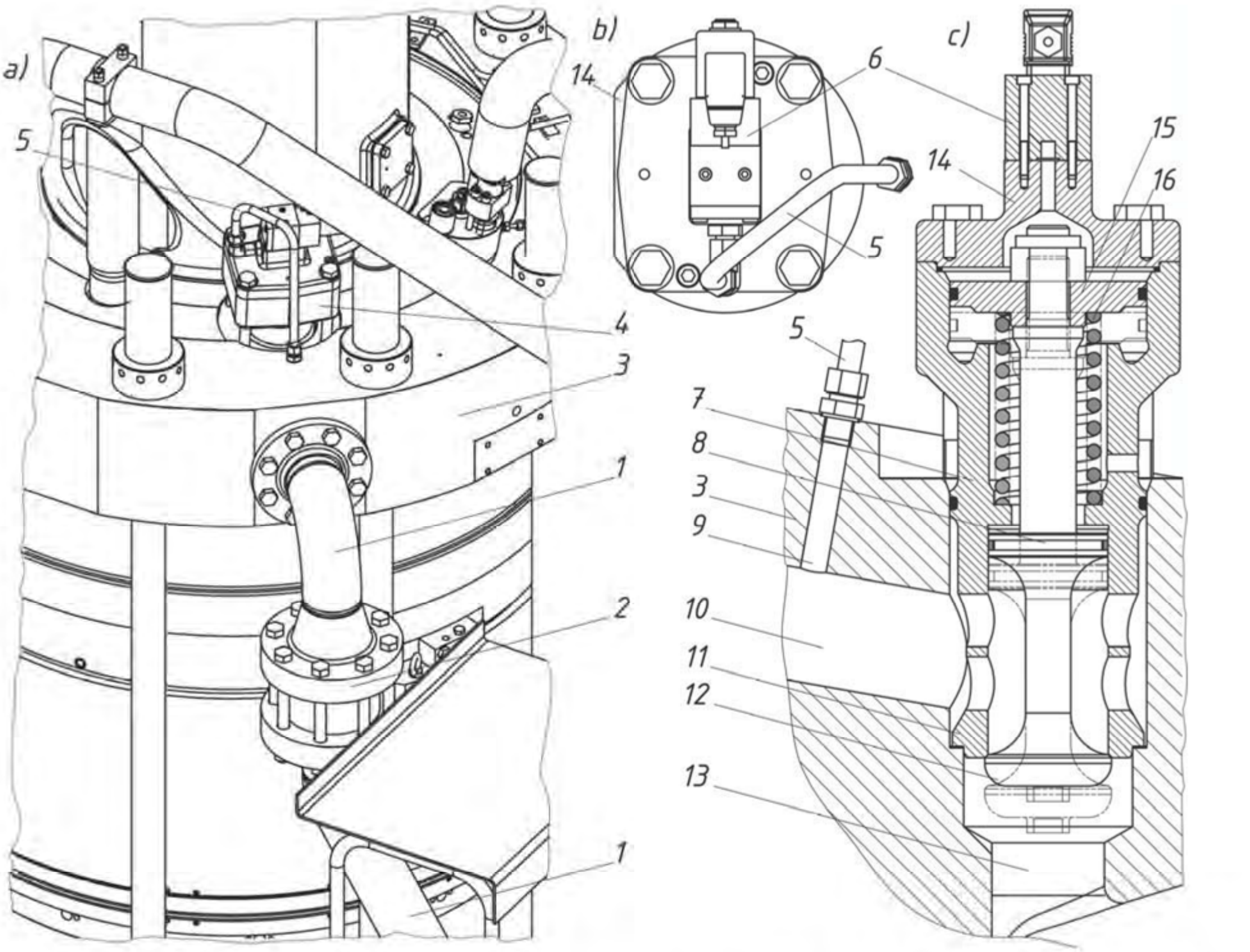


Figure 8.13 – Location of the starting valve (*a*), control valve (*b*) and starting valve (*c*) of low-speed engines of the WX series from WinGD: 1 – air supply line to the starting valve; 2 – flame arrester; 3 – cylinder cover; 4 – start valve; 5 – air supply tube to the control valve; 6 – solenoid valve for controlling the air supply to the cylinder; 7 – starting valve housing; 8 – spindle balancing piston; 9 – air supply channel to the control valve; 10 – air supply channel to the starting valve; 11 – starting valve seat; 12 – starting valve plate; 13 – air supply channel to the working cylinder; 14 – cover; 15 – pneumatic piston for starting valve drive; 16 – return spring (adapted from [14])

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Annex 2

Brief information about scientists and engineers who made a significant contribution to the development and establishment of engine building

Huygens Christian (14.04.1629-08.07.1695) – Dutch mechanic, physicist, mathematician, astronomer and inventor. In 1678 they proposed the first piston engine, which was supposed to use black powder as fuel. The engine proposed by Huygens was never built, but the very idea of burning fuel inside the working cylinder formed the basis for the creation of modern internal combustion engines.

Nicolas Léonard Sadi Carnot (01.06.1796-24.08.1832) – son of the famous politician and mathematician Lazare Carnot. In 1812 He graduated from the Charlemagne Lyceum and entered the Polytechnic School. In 1814 was sent to an engineering school in the city of Metz, and after its completion in 1816 was assigned to an engineering regiment. In 1824 Carnot's first and only work, Reflections on the Motive Force of Fire and on Machines Capable of Developing This Force, was published. This work is considered fundamental in thermodynamics. Died in 1832 from cholera.

Sulzer-Hirzel Johann Jakob (1806-1883) and **Sulzer-Sulzer Salomon** (1809-1869) – founders of the Swiss company Sulzer Brothers. The company began its activities in 1834 from the production of fire pumps, and subsequently switched to the production of ship steam engines. Despite the fact that none of the brothers lived to see the invention of the diesel engine, the innovative spirit inherent in the founders contributed to the fact that everything new found support from the company's management. The brothers' followers realized the promise of diesel engines in time and were among the first to begin producing them for ships. For more than 80 years, the company has been one of the leaders in this industry.

Beau de Rochas Alphonse Eugène (09.04.1815-27.03.1893) French thermodynamic engineer, author of works in various branches of technology. He was involved in the laying of underwater telegraph cables, the organization of railway traffic, and the project of a flooded tunnel across the English Channel. In 1861 developed the theoretical foundations of the four-stroke cycle of an internal combustion engine and substantiated the need for pre-compression as the main condition for increasing the efficiency of the operating process. October 26 1860 received a patent for this cycle, and in 1863 published the book «Recent Research on the Practical Conditions for the Use of Heat» in Paris. Beau de Rocha limited himself only to theoretical work and did not challenge the implementation of his developments by other inventors.

Reithmann Christian (02.09.1818 – 07.01.1909) Austrian watchmaker, self-taught mechanic. For his workshop, he independently built several engines in which the four-stroke cycle was implemented for the first time. October 26 1860 he received the first patent for an engine that had a cylinder diameter 98 mm, piston stroke 111 mm and a rotation speed of 200 min^{-1}. This happened six years before Nichols Ott received a patent for his engine. And although in his patents Reitmann did not defend a four-stroke cycle, but a specific design, he subsequently managed to win a patent dispute against Otto. However, Reitmann retained all rights to the invention for Otto.

Burmeister Karl Christian (1821-1898) and **Wain William** (1819-1882) co-founded the 1846 **Hans Heinrich Baumgarten** of the engineering company, which was later renamed the company Burmeister & Wain. The company was engaged in shipbuilding, having its own shipyards and engineering factories for the production of ship equipment, including steam engines. Subsequently, the company began producing diesel engines based on a patent acquired from R. Diesel. For many years, the company remained a leader in this sector of mechanical engineering, and its research center still operates as part of MAN Diesel SE.

Charles Yale Knight (24.02.1868-05.10.1940) – American self-taught engineer born in state of Indiana. Journalist by training. In 1901, he purchased a car from the already bankrupt company Searchmont. Soon the car's valve spring broke, giving Knight pause. In 1903-1905, he built and tested an experimental four-stroke internal combustion engine, in which gas distribution was not controlled by valves, but by a concentric pair of movable liners inserted into the working cylinder. Knight engines had a number of advantages over valves. Valveless engines had very large intake and exhaust ports, which improved gas exchange. These features made it possible to obtain high powers for those times on Knight engines.

Rudolf Pawlikowski (16.06.1868-10.11.1942) German engineer. In 1893 graduated from the Dresden Polytechnic University, after which he worked with H. Junkers and O. Miller. With 1897 worked together with R. Diesel on his «rational» engine. In 1898 he moved to Dessau, where he worked as chief engineer at Görlitzer Maschinenbau AG, and in 1902 took up a position as a civil engineer in Görlitz. The main work was devoted to the technical implementation of an engine running on coal dust. In total, more than 200 patents were received. The first workable engine was built in 1916, which the inventor, by analogy with a diesel engine, called the Rupa-motor (RuPa).

Lanchester Frederick (23.10.1868-08.04.1946) – English engineer and inventor who made a significant contribution to the development of various branches of technology, including internal combustion engines. He studied at a number of leading technical educational institutions in England, but never received a full education. He was one of the founders of the automobile manufacturing company Lanchester Motor Company, Ltd. In the field of internal combustion engines, he developed a surface carburetor (1895), a torsional vibration damper (1910), a method of balancing a single-cylinder engine (1911), etc.

Grinevetsky Vasily Ignatievich (02.06.1871-31.03.1919) was born in Kiev, from where the family moved to Kazan, where he graduated from 1889 real school. In 1889 becomes a student at the Imperial Moscow Technical School (IMTU), which he graduates from 1896 and stays to teach. In 1902 Grinevetsky became a professor. IN 1914 he is appointed director of IMTU. Grinevetsky was one of the first in Russia to understand the prom-ise of internal combustion engines and began to seriously study them. In 1907 The first edition of G. Guldner's book «Gas, Oil and Other Engines» was published. In the appendix to the translation of the book, Grinevetsky included his work «Thermal Calculation of the Work Process», which laid the foundations for theoretical approaches to the design of internal com-bustion engines.

Zeiliger Miron Pavlovich (1874-1952) – heating engineer at the Ludwig Nobel plant. He received his education at the St. Petersburg Institute of Technology, and then taught there. There is very little information about Zeiliger's work in Russia, as well as information about himself. After the revolution, in 1924 he immigrated to Paris, where he worked as a professor at the Russian Higher Technical Institute and lectured on internal combustion engines and thermodynamics. He was a member of the board of the Society of Russian Engineers, where he lectured. The main works on calculating the operating process of high-speed engines are devoted to comparing the thermal cycles of mixed heat supply and the isobaric cycle of Diesel.

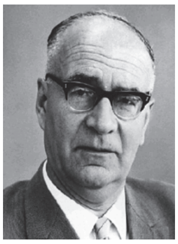
L'Orange Prosper (01.02.1876-30.07.1939) – German engineer and inventor, studied at the Technical University of Berlin Charlottenburg. After graduation he worked at Gasmotorenfabrik Deutz, and then 1910 rhe joined Benz & Cie, where he headed the production of stationary engines. In 1908 patented a high-pressure fuel injection pump with spool-type control of cyclic flow and a needle valve for closed-type injectors, and in 1909 he proposed the use of a preliminary chamber (pre-chamber) in diesel engines, which made it possible to create compact high-speed engines for ground transport. Together with his sons, he founded the L'Orange company in 1933, which today remains the world leader in the production of fuel equipment for diesel and gasoline engines.

Trinkler Gustav Vasilievich (24.04.1876-04.02.1957) – Russian scientist and inventor, creator of a compressorless engine, the first design of which he presented in 1899 as my graduation project. In the spring 1902 engine production was established at the Kerting brothers plant in Hannover. Upon returning to Russia, Trinkler spent a number of years working at the Sormovo plant, improving the design of his engine. Since autumn 1917 began his teaching career. In 1930 without defending a dissertation, he is awarded the academic degree of Doctor of Technical Sciences. Until the end of his life he maintained close contact with the Sormovo plant.

Clessie Lyle Cummins (27.12.1888 – 17.08.1968) – founder of Cummins Engine Co. Became the first American to build and install diesel engines in trucks, buses and cars. He received more than 30 patents for inventions, including many fuel injection systems for his engines. By age 12, Cummins had built a working steam engine. In 1919, he founded the Cummins Engine Company and began producing marine diesel engines. Ten years later, Cummins installed the first diesel engine in a car, which he drove for promotional purposes 800 miles to the New York Auto Show for only $1.38 in fuel. The following year, a Cummins diesel car took part in the Indianapolis 500, finishing 12th and driving the entire track without refueling.

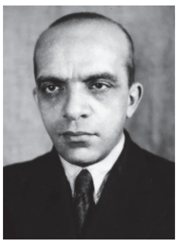
Pielstick Gustav (25.01.1890-11.03.1961) – German engineer. Graduated from the Kiel Higher School of Shipbuilding and Mechanical Engineering. In 1911 began working at MAN as a design engineer for submarine engines. During World War I he became chief engineer, and after the war he developed four-stroke engines for commercial shipping. In 1934 become director of the MAN diesel engines department. He paid special attention to the development of high-power turbocharged engines. With 1946 moved to France, where he founded the diesel engine design bureau Société d'Etudes des Machines Thermiques (SEMT). Pielstick's work as director was so successful that the company was subsequently named SEMT Pielstick.

Vanscheidt Vsevolod Aleksandrovich (07.08.1890-27.09.1982) graduated from the mechanical department of the St. Petersburg Technological Institute in 1914 Worked at the «L. Nobel» (now «Russian Diesel»). Under his leadership in 1925 designed and built the country's first 2-stroke non-compressor engine with a power of about 37 kW. With 1929 conducted teaching work at the shipbuilding department of the Leningrad Polytechnic Institute, and in 1930 1971 Head of the Department of Internal Combustion Engines at the Leningrad Shipbuilding Institute, professor. In 1938 And 1950 His textbooks on the theory of marine internal combustion engines were published.



Wankel Felix Heinrich (13.08.1902-09.10.1988) – German self-taught inventor. IN 1924 he came up with the idea of creating a rotary piston engine, which he worked on all his life. It was only possible to build a working engine until 1957, after engineer Walter Freude joined the work. To create a functional engine, Wankel carried out extensive research and formulated requirements for effective moving seals. Despite the lack of special education, in 1969 Wankel becomes a doctor of technical sciences at the Munich Institute, and then a professor.

Vibe Ivan Ivanovich (27.12.1902-27.12.1969) – scientist, specialist in the field of internal combustion engine theory, Doctor of Technical Sciences. In 1926 Graduated from Zaporozhye Mechanical Engineering Institute. In 1932 he graduated from graduate school at the Leningrad Civil Aviation Training Center. In 1938-41.Head of the Department of Special Engines at the Stalingrad Mechanical Institute. With 1965 Professor of the Chelyabinsk Polytechnic Institute. Author of numerous works on the theory of internal combustion engines. The formula he proposed for calculating combustion processes has become widespread in all countries of the world.



Glagolev Nikolai Matveevich (1903-20.04.1976) scientist in the field of theory and design of internal combustion engines. From 1954 to 1970 Head of the department of internal combustion engines at the Kharkov Polytechnic Institute. He made significant efforts to develop the scientific school of engine building in Kharkov and to its recognition both in the country and abroad. In 1949 he defended his doctoral dissertation on the topic «A new method for calculating the working processes of internal combustion engines». In 1950 he was awarded the title of professor. Author of 11 books and more than 100 works in the field of internal combustion engines.

Fomin Yuri Yakovlevich (29.06.1926-05.07.2000) graduated with honors from the ship mechanics department of the Vladivostok Higher Marine Engineering School in 1950 and was soon enrolled in graduate school at the Leningrad Higher Marine Engineering School. After graduating from graduate school, he was sent to the Odessa Institute of Marine Engineers, where he worked his way up from assistant to department head. In October 1965, he defended his doctoral dissertation on the topic «General method of hydraulic calculation of marine diesel fuel systems». All scientific activities of Fomin Yu. was devoted to the research and development of methods for calculating fuel systems of marine diesel engines. He published about 340 scientific works, including 9 monographs and 15 brochures.

Навчальне видання

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